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XXI.

AN ANALYSIS OF THE RESULTS OF THE EXPERIMENTS MADE ON THE PADDLE-WHEEL STEAMER, *VILLE DE DOUVRES*, BY THE RESEARCH COMMITTEE ON MARINE ENGINE TRIALS OF THE INSTITUTION OF MECHANICAL ENGINEERS.

By Chief Engineer Isherwood, U. S. Navy.

The Committee of Research on Marine Engine Trials of the Institution of Mechanical Engineers (England) has reported, by Professor Alexander B. W. Kennedy, the data and results of its experiment made on the 8th of September, 1891, with the machinery of the paddle-wheel steamer, *Ville de Douvres*, belonging to the Belgian postal service between the cities of Ostend and Dover.

The report has been published, *in extenso*, in the last August number, page 408 of this journal, but as it gives only the bare observed data, without expression of opinion, or tracing of inferences, or drawing of general deductions, only a portion of the benefit capable of being derived from it will be had, unless the reader

takes the data and makes from it another report supplying these omissions. This the writer has done according to his views, and, believing they may be as useful to others as to him, contributes them to the JOURNAL.

The vessel and machinery were constructed by the Cockerill Company at Seraing, Belgium, and completed in February, 1890. The vessel had made 220 passages when experimented with, and the machinery was in its ordinary running condition.

The data of the experiment allows the determination, for the experimental conditions, of the speed of the vessel, the performance of the paddle wheels, the power developed by the engine, the weights of fuel and of feed water consumed per unit of time, the economic development of the power relatively to fuel, heat and water consumed, the difference between the weight of water pumped into the boilers and the weight of steam accounted for by the indicator, the feed water consumed per unit of time by the auxiliary engines (the power developed by them was not ascertained), the chemical composition of the coal consumed and its evaporative efficiency in the particular kind of boiler; also the chemical composition of the gases of combustion, and the distribution of the heat of the fuel. The only item of importance omitted as regards the motive engine, and the omission is a very important one, was the determination of the weight of injection water, which quantity allows, supposing all the quantities involved to be exact, an appreciation of the quantity of heat required to perform the molecular work of the steam; that is to say, the heat transmuted into the work of separating the molecules of steam farther apart during its expansion or of overcoming its cohesion. Of course, owing to the great quantity of injection water used per hour, its measurement, even by meters, would have been a labor of some difficulty, but by using a number of large meters it could have been done. Individuals making experiments similar to this, with few facilities and slight resources, are not expected to attain completeness; but when an experiment is made by a British "Committee of Research on the Steam Engine," with apparently abundant means and no facility lacking, the engineering public expects much more, and that

important facts will be determined which, from the difficulties and cost involved, can scarcely be asked from individual experimenters.

The temperature of the gases of combustion in the chimney was too high to be taken by the mercurial thermometers of the committee, although they had compressed nitrogen above the mercury. They broke at the temperature of 860 degrees Fahrenheit. The committee estimated the temperature at 910 degrees. Probably 950 degrees, which is the temperature the writer will use hereinafter, is nearer the truth. No provision was made for so probable an accident, nor was the temperature of the air in the boiler rooms ascertained. The report is also deficient in not giving all the elements of the vessel.

The indicators were standardized, the coal was carefully weighed on tested scales, the feed water was measured by a tested water meter specially constructed for the occasion, the boiler steam was tested for priming, and every precaution was taken to ensure accuracy of all the observations. Every facility was furnished to the committee. The vessel was run over a specially chosen route—not in its regular service—in perfectly smooth water and in fine weather.

The indicator diagrams were taken by separate instruments applied to each end of each cylinder, 78 single diagrams in all being taken at equal intervals for each set of 4 diagrams, during the run of nine consecutive hours, and from the mean of them was obtained the mean pressures hereinafter used.

The safety valves on the boilers were set to the pressure 118 pounds per square inch above the atmosphere.

The two fire rooms were closed air tight, and the air for combustion was supplied to them by two centrifugal blowers, one to each fire room, having blades of 60 inches diameter and 20½ inches width, each driven by a Brotherhood engine of three single-acting cylinders of 5 inches diameter with 5 inches stroke of piston. The air pressure in the fire rooms varied from that due to a column of water 1.0 to 0.7 inch high, mean 0.85 inch. The minus pressure in the chimneys, shown by difference of water level in the legs of a U-tube, was equivalent to that of a column

of water 0.22 inch high, making the chimney draft equivalent to that due to a pressure equal to the pressure of a column of water (0.85 + 0.22 =) 1.07 inches high. During the experiment the blowers filled the fire rooms with coal dust.

The number of revolutions made by the paddle wheels was taken by a counter every half hour. The regularity of the performance may be judged from the facts that the mean number made per minute during the half hour giving the maximum number was 38.1, and during the half hour giving the minimum number was 36.4.

As there was no possibility of separating during the experiment the feed water consumed by the main engine from the feed water consumed by the auxiliary engines, the aggregate quantity being measured simultaneously by the same meter, the feed water consumed per hour by each of the auxiliary engines had to be ascertained by separate trials, which were made after the termination of the experiment. The trial of each auxiliary engine (the one supplying the circulating or injection water to the condenser of the engine, and the two driving the two blowers supplying air to the fire rooms) lasted two consecutive hours, during which it was operated at the same number of revolutions per minute and with the same boiler pressure as during the experiment. One of the boilers was used for this purpose, and the water was run up in it sufficiently high to outlast the trial. The height between the two water levels, at the beginning and end of the trial, was taken and this height was then refilled by the meter.

The feed water entered the boiler with the temperature at which it left the condenser. There was neither feed-water heating nor steam superheating.

During the experiment the exhaust steam from the auxiliary engines was discharged directly into the condenser.

HULL.

The hull was of steel, with its bottom painted and clean. The water lines were excessively sharp, the vessel being designed wholly for speed. The coefficient of fineness, relatively to the circumscribing parallelopipedon, was about 0.5.

Length of the hull on the water line, in feet		271
Extreme breadth of the hull on the water line, in feet		29
Moulded depth of the hull, in feet		15½
Draught of water during the trial, in feet and inches	Forward	8′ 6½″
	Mean	9′ 0¾″
	Aft	9′ 7″
Displacement in tons at the above draught of water		1,090

ENGINE.

One inclined, surface-condensing, compound engine, the two cylinders of which lie side by side. The cylinders are not steam jacketed, but the receiver surrounds the small cylinder concentrically. Each cylinder has one piston rod, but no tail rod. The small cylinder has two piston valves, and the large cylinder has one piston valve. These valves are operated by the Allan link, and they cut off the steam in both cylinders by lap at from 50 to 75 per centum of the stroke of the pistons.

There are two air pumps, two feed pumps and two bilge pumps, all vertical and single-acting. One of each is driven from each of the two main cross-heads by bell-crank levers.

The surface condenser occupies the space beneath the guides of the two main crossheads and supports them. The condenser tubes are arranged in two groups in the same shell, with an 18 inches wide water space vertically between them. The circulating or injection water enters at the bottom and outer side of each group, passes three times through it, and discharges into the top of the 18 inches wide water space, whence the discharges for both groups go overboard. The exhaust steam from the large cylinder is equally divided, and by means of two branch pipes is delivered equally over the tops of both groups of tubes.

There was one centrifugal circulating pump to supply the condenser with injection water. This pump was 2 feet and 10 inches in diameter over the blades, and was driven by a single double-acting steam cylinder 10 inches in diameter with 10 inches stroke of piston.

The following are the principal dimensions of the engine:

Number of small cylinders	1.
Diameter of small cylinder, by gauge	50.12 inches.
Diameter of piston rod of small cylinder	8.27 inches.

Net area of the piston of the small cylinder, exclusive of rod.. 1946.0782 square inches.
Stroke of the piston of the small cylinder..... 6 feet.
Space displacement of the piston of the small cylinder, per stroke, 81.0866 cubic feet.
Waste space in clearance, etc., at one end of the small cylinder, 12.1630 cubic feet.
Fraction which the waste space at one end of the small cylinder is of the space displacement of its piston per stroke..... 0.15
Number of large cylinders..... 1.
Diameter of large cylinder, by gauge..... 97.12 inches.
Diameter of piston rod of large cylinder..... 8.27 inches.
Net area of the piston of the large cylinder, exclusive of rod, 7381.2661 square inches.
Stroke of the piston of the large cylinder..... 6 feet.
Space displacement of the piston of the large cylinder per stroke, 307.5527 cubic feet.
Waste space in clearance, etc., at one end of the large cylinder... 36.9063 cubic feet.
Fraction which the waste space at one end of the large cylinder is of the space displacement of its piston per stroke..... 0.12
Ratio of the net area of the piston of the large cylinder to the net area of the piston of the small cylinder taken as unity..... 3.7929
Ratio of the space displacement of the piston of the large cylinder per stroke plus the waste space at one end of that cylinder, to the space displacement of the piston of the small cylinder per stroke plus the waste space at one end of that cylinder, the aggregate taken as unity..... 3.6939
Number of tubes in the surface condenser (brass)..... 6,026
Outside diameter of the tubes..... 0.79 inch.
Length of tubes between tube plates..... 5.25 feet.
Aggregate area of outside surface of tubes..... 6,543.1242 square feet.
Number of air pumps (single acting)..... 2.
Diameter of air pumps..... 32.68 inches.
Stroke of air-pumps pistons..... 24.61 inches.
Space displacement of the piston of each air pump, per stroke..... 11.9460 cubic feet.
Number of feed pumps (single acting)..... 2.
Diameter of feed pumps..... 6.69 inches.
Stroke of feed-pump plungers..... 24.61 inches.
Space displacement of the plunger of each feed pump, per stroke.. 0.5006 cubic feet.
Number of bilge pumps (single acting)..... 2.
Diameter of bilge pumps..... 6.69 inches.
Stroke of bilge-pump plungers..... 24.61 inches.
Space displacement of the plunger of each bilge pump, per stroke..0.5006 cubic foot.

PADDLE WHEELS.

The paddle wheels have feathering paddles entering, passing through and leaving the water in nearly vertical positions. They are of steel and curved in the vertical direction. The concavity of those in the water faces aft. The following are the principal dimensions:

Diameter of the wheels over the paddles ..... 22 feet 10 inches.
Number of paddles in each wheel ..... 9.
Length of each paddle ..... 10 feet.
Breadth of each paddle ..... 4 feet 4 inches.
Area of each paddle ..... 43⅓ square feet.
Diameter of the axis of the paddles ..... 19 feet 2 inches.
Deepest immersion of outer edge of paddles ..... 5 feet 9 inches.

BOILERS.

There are four single-ended cylindrical boilers, with return-tubes above the furnaces. Two of the boilers are placed forward and two abaft of the engine, in separate fire rooms. Each two have one chimney in common, making two chimnies in all. Each boiler has three cylindrical plain (non-corrugated) furnaces, and each furnace has 66 wrought-iron grate bars.

The boilers are worked with a mechanical draft produced by an air pressure above the atmospheric pressure in the air-tight fire rooms. The pressure is maintained by one centrifugal blower of 60 inches diameter and 20¼ inches breadth of blades for each fire room. This blower is worked by a Brotherhood engine of three single acting cylinders, 5 inches in diameter, with 5 inches stroke of piston.

The following are the principal dimensions and proportions of the boilers:

Number of boilers ..... 4.
Diameter of boiler shell ..... 13 feet.
Length of boiler shell ..... 10 feet.
Number of furnaces ..... 12.
Diameter of the furnaces ..... 3 feet.
Length of the grates ..... 6 feet 6½ inches.
Area of grate surface ..... 235.5 square feet.
Number of "expanded in" tubes ..... 1,044.
Number of "stay" tubes ..... 312.
Total number of tubes ..... 1,356.
Outside diameter of all the tubes ..... 2.48 inches.
Inside diameter of the "expanded in" tubes ..... 2.24 inches.
Inside diameter of the stay tubes ..... 1.93 inches.
Length between plates of all the tubes ..... 7 feet 1⅛ inches.
Heating surface in all the tubes, calculated for outside diameter, 6,282.035 square feet.
Heating surface in all the tubes, calculated for inside diameter, 5,493.418 square feet.
All other heating surface in the boilers ..... 1,060.000 square feet.

Total heating surface in the boilers, calculated for the inside diameter of the tubes	6,553.418 square feet.
Cross area of the tubes for draft	34.9097 square feet.
Number of chimnies	2.
Diameter of chimnies	5 feet 3 inches.
Aggregate cross area of the two chimnies	43.295 square feet.
Height of chimnies above level of grates of lower furnaces	49 feet 8 inches.
Ratio of the heating to the grate surface, (inside of tubes)	27.8277
Ratio of the grate surface to the cross area of the tubes	6.7460
Ratio of the grate surface to the cross area of the chimnies	4.7773

WEIGHTS OF MACHINERY.

Main engine	145.27 tons.
Circulating pump	2.20 tons.
Donkey pump	1.74 tons.
Pipes, valves, etc	10.37 tons.
Paddle wheels	45.21 tons.
Boilers	132.35 tons.
Water in boilers	61.51 tons.
Water in condenser, etc	7.85 tons.
	406.50 tons.
Add for paint, putty, gaskets and lead for joints, lagging, non-heat-conducting coverings, blowing engines, etc., etc	5.50 tons.
	412.00 tons.

COAL.

The coal consumed was "briquettes" from the factory of Messrs. Dehaynin, of Marcinelle. It was analyzed by Mr. C. J. Wilson, and had the following chemical composition centesimally:

	Crude coal.	Coal exclusive of ash and water.
Carbon	84.65	91.722
Hydrogen	3.98	4.312
Nitrogen, Sulphur, Oxygen,	3.66	3.966
Ash	5.30	0.000
Water	2.41	0.000
	100.00	100.000

As a steam generating coal in every respect, the above has no superior.

The incineration of the refuse drawn from the ashpits gave the following:

Mineral matter	57.14 per centum.
Combustible matter	42.86 per centum.
	100.00 per centum.

The experimental refuse was 7.17 per centum of the crude coal, and as 57.14 per centum of it was mineral matter, the true ash for the coal consumed during the experiment would be $(7.17 \times 0.5714 =)$ 4.097 per centum of the crude coal, or about $1\frac{1}{6}$ per centum less than given by direct analysis. Some of this difference, probably, was swept out of the chimnies by the force of the powerful draft.

According to the experiments made by Scheurer-Kestner on Anzin steam coal (the variety used), the total heat of combustion of one pound of what remained, after deduction of hygroscopic water and ash, was 16,662.6 Fahrenheit units.

As the ash and water combined was 7.71 per centum of the coal, the pound of crude coal had a total heat of combustion of 15,377.9 Fahrenheit units.

Also the pound of what in this paper is called "combustible," that is, the crude coal exclusive of the 5.30 per centum of ash, but inclusive of the 2.41 per centum of hygroscopic water, had the total heat of combustion of 16,227.5 Fahrenheit units.

The chemical composition of the pound of "combustible" is, centesimally, as follows:

	Coal exclusive of ash but inclusive of water.
Carbon	89.38754
Hydrogen	4.20274
Nitrogen, Sulphur, Oxygen,	3.86484
Water	2.54488
	100.00000

GASES OF COMBUSTION.

Mr. Wilson also analyzed the gases of combustion from the above coal. At intervals of one hour samples were drawn from

both chimnies and analyzed separately. There were thus nine samples from each chimney, or eighteen samples in all, and the results are as follows, centesimally, for the *dry* gases, *by volume:*

	Forward chimney.	Aft chimney.	Mean from both chimnies.
Carbonic acid	10.81	12.30	11.55
Carbonic oxide	0.00	0.00	0.00
Oxygen	8.72	7.18	7.95
Nitrogen	80 47	80.52	80.50
	100.00	100.00	100.00

In the above analyses, no attempt was made to ascertain whether either uncombined hydrogen, or carburetted hydrogen, was among the gases of combustion. As there was no carbonic oxide, there was probably no other combustible gas present. Almost exactly the same weight of coal being consumed in the forward and in the aft boilers, the mean composition of the chimney gases from the two can be accepted.

DISTRIBUTION OF THE TOTAL HEAT OF COMBUSTION OF THE COAL.

For the purpose of determining the distribution of the total heat of combustion of the coal, the experimental data is unfortunately incomplete in the following five items, namely: The temperature of the air in the fire rooms, the temperature of the gases of combustion in the chimnies, and the per centum of nitrogen, sulphur and oxygen in the coal. The best that can be done is to supply these deficiencies with reasonable approximations. The temperature of the air in the fire rooms supplying the oxygen for combustion will be taken at 100 degrees Fahrenheit, and the temperature of the gases of combustion in the chimney will be taken at 950 degrees Fahrenheit. Mr. Wilson's analysis of the coal gives the *aggregate* per centum of the nitrogen, sulphur and oxygen in the "combustible" part of the crude coal at 3.86484, separating which in the usual proportions for such coal, the nitrogen may be taken at 1 per centum, the sulphur at 1 per centum, and the remaining oxygen at the remaining 1.86484 per centum. The greatest error that could occur in this apportionment would not practically affect the result.

Errors in the temperature of the air in the fire rooms and in the chimnies affect the quantity of heat contained by the gases of combustion in the chimnies, and the quantity of heat transmuted into the work of displacing the atmosphere by them. Indirectly, these errors affect the determination of the quantity of heat radiated from the boilers, which is by difference.

The combustion being "perfect," that is the analyses showing no combustible gases in the chimney, the calculations become very simple. The first quantity to be ascertained is the number of pounds of oxygen chemically required to saturate the carbon, hydrogen and sulphur of one pound of the "combustible" part of the crude coal.

	Constituents in fractions of a pound, of one pound of "combustible."		Pounds of oxygen required chemically per pound of "combustible."
Carbon	0.8938754	$\times 2\frac{8}{3} =$	2.3836678
Hydrogen	0.0420274	$\times 8 =$	0.3362192
Sulphur	0.0100000	$\times 1 =$	0.0100000
Oxygen	0.0186484		
Nitrogen	0.0100000		
Hygroscopic water	0.0254488		
	1.0000000		2.7298870

The free oxygen in the chimney gases being by volume 7.95 per centum of the dry gases of combustion, only (20.96 − 7.95 =) 13.01 were burned; consequently the oxygen supplied was $\left(\frac{20.96}{13.01} =\right)$ 1.611068 times the quantity chemically required, or (2.729887 × 1.611068 =) 4.3980336 pounds per pound of "combustible," deducting from which the 0.0186484 pound of oxygen in the pound of "combustible," there remain 4.3793852 pounds of oxygen to be obtained from the atmosphere. To supply this oxygen there are required of natural air at the temperature of 64 degrees Fahrenheit and three-fourths saturated with aqueous vapor, 252.60260473 cubic feet, weighing 19.0344387 pounds. Of this weight of air (2.7298870 − 0.0186484 =) 2.7112386 pounds of oxygen combined with the pound of "combustible," leaving (19.0344387 − 2.7112386 =) 16.3232001 pounds as excess of air, and as nitrogen corresponding to the 2.7112386

pounds of oxygen. This nitrogen was (2.7112386 × 3.347826 =) 9.0767551 pounds, leaving (16.3232001 − 9.0767551 =) 7.246-4450 pounds of unchanged air.

The pounds of the products of combustion due to the combustion of the constituents of one pound of combustible, multiplied by their respective specific heats, are as follows:

Gases of combustion.	Weights in pounds of the constituents of the gases of combustion per pound of "combustible."		Specific heat.		Fahrenheit units of heat per degree of temperature, in the constituents of the gases of combustion of one pound of "combustible."
Carbonic acid	3.2775432	×	.2169	=	0.710899120
Water of hydrogen combination,	0.3782466	×	.4805	=	0.181747491
Hygroscopic water	0.0254488	×	.4805	=	0.012228148
Sulphurous acid	0.0200000	×	.1544	=	0.003088000
Nitrogen of the combustible.	0.0100000	×	.2438	=	0.002438000
Nitrogen of air due to the combined oxygen of the air,	9.0767551	×	.2438	=	2.212912893
Air in excess	7.2464450	×	.239258	=	1.733769938
	20.0344387				4.857083590
			Mean specific heat =		0.24243672

The bulk of the gases of combustion in the chimnies at the temperature of 950 degrees Fahrenheit is calculated as follows: The number of cubic feet of carbonic acid, sulphurous acid, nitrogen, and air per pound weight—these gases being at the temperature of 32 degrees Fahrenheit and under the atmospheric pressure—are known, and they constitute the dry gases of combustion in the present case; the steam additionally mingled with them is that which is derived from the hygroscopic water of the "combustible," and from the combination of the hydrogen of the combustible with oxygen.

Having ascertained the aggregate number of cubic feet of the *dry* gases of combustion per pound of "combustible" at the temperature 32 degrees and under the atmospheric pressure, this aggregate must be multiplied by the heat coefficient of expansion of these gases for the temperature interval between 32 and 950 degrees. The coefficient is assumed to be the same for all the gases at the high temperature involved, and to be that of an

ideal gas, namely, 0.3662 for a temperature interval of 180 degrees.

The number of cubic feet of saturated steam at the temperature of 212 degrees and under the atmospheric pressure, being known per pound weight (26.31087 cubic feet), the fraction of a pound weight of steam per pound of "combustible" must be multiplied by these cubic feet and by the coefficient of expansion for the temperature interval between 212 and 950 degrees, the same coefficient being used as before, and the product is to be added to the final product previously obtained for the dry gases; the sum will be the number of cubic feet of the gases of combustion at the chimney temperature 950 degrees, for one pound of "combustible." For example:

Dry gases in the gases of combustion.	Pounds of dry gases per pound of "combustible."		Cubic feet per pound at 32° under atmospheric pressure.		Cubic feet of dry gases of combustion at 32° and under atmospheric pressure per pound of "combustible."
Carbonic acid.....	3.2775432	×	8.1011782	=	26.551961521398
Sulphurous acid..	0.0200000	×	5.6000750	=	.112001500000
Nitrogen	9.0867551	×	12.7524298	=	115.878206522542
Unchanged air...	7.2464450	×	11.5779586	=	83.899040207177
			Aggregate	=	226.441209751117
			Multiply by coefficient of expansion.....		2.86762
			Cubic feet of dry gases at 950°.....	=	649.347341906498
			Add cubic feet of steam at 950°.....	=	26.569025639624
			Cubic feet of gases of combustion at 950°	=	675.916367546122

In the following calculations the necessary experimental quantities are taken from the table, hereinafter given, containing the data and results of the trial.

The difference between the temperature of the "combustible" when thrown into the furnace, and of the gases of combustion when leaving the boiler, being (950° — 100° =) 850° Fahrenheit, the number of Fahrenheit units of heat imparted in the boiler to the gases of combustion when leaving it, are (4.85708359 × 850 =) 4,128.521051.

The pound of "combustible" during the experiment vaporized in the boiler ($\frac{595.620}{61.660}$ =) 9.659747 pounds of water from the temperature 158° Fahrenheit and under the pressure of 120.63 pounds per square inch above zero, which is equivalent to

(9.659747 × 1,059.53057 =) 10,234.797245 Fahrenheit units of heat.

The number of Fahrenheit units of heat required to vaporize the hygroscopic water in the pound of "combustible" from the temperature of 100 degrees Fahrenheit, is (0.0254488 × 1,078.-6557 = 27.450493.

The number of Fahrenheit units of heat transmuted into the work of the displacement of the atmosphere by the difference between the volume of the air entering the furnace (252.60260-473 × 1.07334 = 271.128479760898 cubic feet, allowing for the expansion from the temperature of 64° to that of 100°,) and the volume of the gases of combustion when leaving the boiler (675.916367546122 cubic feet), is calculated as follows:

675.916367546122 — 271.128479760898 × 2115 ÷ 789.25 = 1084.734093.

The sum of the above four quantities is 15475.502882 Fahrenheit units of heat, and it includes the whole of the total heat of combustion with the exception of the heat lost by radiation from the boiler surfaces; consequently the difference between that sum and the total heat of combustion will be the quantity of heat lost by radiation. The total heat of combustion of one pound of "combustible" being 16227.5 Fahrenheit units, the heat lost by radiation will be, by difference, (16227.5 — 15475.502882 =) 751.997188 Fahrenheit units.

As long as steam of the same pressure is used in the boiler, and with the air in the fire room at also the same temperature, this latter quantity of heat will be constant, let the rate of combustion or quantity of vaporization per hour be what they may. When, by lessened rate of combustion, the temperature of the gases of combustion in the chimney is lessened and the quantity of heat carried off in them is proportionally lessened, and further, the quantity of heat transmuted into the work of displacing the atmosphere by the escaping gases is also correspondingly lessened, the quantity of heat expended in vaporizing the water in the boiler will be increased by exactly the quantity that the two previous items have been diminished. In other words, other things being equal, the quantity of heat expended in vaporizing

the hygroscopic water of the pound of coal is constant, the heat lost by radiation per pound of coal is constant, and the sum of the heats expended in the gases of combustion leaving the chimney, in the displacement of the atmosphere by these gases and in the vaporization of water in the boiler, is constant per pound of coal, let the rate of combustion be what it may. The distribution of the constant quantity of heat consisting of these three items will, of course, be different at different rates of combustion, the last item becoming less and the two others becoming greater as the rate of combustion is increased, the decrease in the one exactly equaling the increase in the other two. Hence, per pound of coal, the percentage of loss of heat by radiation will always be the same as well as that loss in Fahrenheit units, other things remaining the same. But, as the loss of heat by radiation *per unit of time* (not *per unit of weight of coal*) remains constant, let the rate of combustion be what it may, while the quantity of heat generated per unit of time is proportional to the rate of combustion, evidently the greater the weight of coal consumed per hour in a boiler, the less *proportionally* will be the loss of heat by radiation. Further, as with greater rates of combustion the percentage of the heat of combustion in the water vaporized in the boiler becomes less, while the percentage of heat lost by radiation continues constant, the latter loss increases *relatively* to the heat utilized in vaporization, in proportion as the rate of combustion increases.

The following collection shows the total heat of combustion, and its distribution, of one pound of "combustible," the combustible being what remains of the crude coal after deduction of its ash, but including its hygroscopic water:

	Heat of combustion of one pound of the "combustible" in Fahrenheit units.	Per centum of the total heat of the combustion of the pound of "combustible."
Heat imparted to the water vaporized in the boilers.....	10,234.797245	63.07070
Heat in the gases of combustion leaving the boiler above the temperature of the air in the fire rooms....	4,128.521051	25.44151

Heat transmuted into the work of displacing the atmosphere by the gases of combustion leaving the boilers...	1,084.734093	6.68454
Heat vaporizing the hygroscopic water of the "combustible".....	27.450493	0.16916
Heat lost by radiation from the surfaces of the boilers.....	751.997118	4.63409
Total heat of combustion.....	16,227.500000	100.00000

The quantity of heat radiated must, of course, be expressed in relation to time, and as in the present case there were consumed 6851.111111 pounds of combustible per hour, there were radiated ($751.997118 \times 6851.111111 =$) 5152015.810653 Fahrenheit units of heat per hour. This radiation was from the external surfaces of the boilers, including uptakes and chimnies.

As the aggregate cross area of the boiler tubes for draft was 34.9097 square feet, and as the volume of the gases of combustion entering the chimney per hour was ($675.916367546122 \times 6851\frac{1}{9} =$) 4630778.135877098058 cubic feet, the velocity of the gases entering the chimnies was 36.847273 feet per second. This calculation, of course, supposes that the gases were equally distributed in the tubes, which may not have been the fact.

TABULATED DATA AND RESULTS OF THE EXPERIMENT.

In the following table will be found the data and results of the experiment grouped for facility of reference. Each quantity is so fully described on its respective line that but little additional explanation is required.

On the last two lines, under the head of "Total Quantities," will be found the per centum of the total weight of feed water pumped into the boilers converted into steam for use in the auxiliary engines. The per centum for the auxiliary engine driving the circulating pump was 3.127833, and the per centum for the auxiliary engines driving the centrifugal blowers was 3.445150, total 6.572983 per centum, which, however, does not represent the per centum of the aggregate indicated horses-power developed by these engines and by the main engine, developed by the auxiliary engines, because the cost of the unit of the auxiliary power in weight of steam consumed per unit of time is probably

nearly three times the similar cost of the unit of power developed by the main engine. In the present case the auxiliary power was probably not more than 2½ per centum of the aggregate auxiliary and main engine power.

The "vacuum" in the condenser, given under the head of "Engine," was only 20.6 inches of mercury, equivalent to a pressure in the condenser of 4.72 pounds per square inch. Now, under the head of "Temperatures," the temperature of the feed water is given at 158 degrees Fahrenheit. The temperature of steam of the above condenser pressure is 159.92 degrees, and, as the condenser temperature must necessarily be somewhat higher than the hot-well temperature, the inference is warranted that there was no air inleakage. The temperature of the discharge water was only 85 degrees; that of the injection water being 61.7 degrees. The difference between the temperatures of the feed water and the discharge water is thus enormous, being (158 — 85 =) 73 degrees, showing the inefficiency of the condensing surface in transmitting the heat from the steam on one side of it to the injection water on the other side. This inefficiency could be due but to one cause, namely, the deposition of some non-heat conducting material on the condensing surfaces. In the present case the material was probably mud, for the vessel at the Belgian end of its voyages runs for a considerable distance in the water of an extremely muddy river. This mud deposits upon the inside of the condensing tubes, concentrically, leaving only the small diameter open required for the passage of the injection water under the pressure of the pump; and it bakes very hard upon the tube surface, for the high temperature is long continued, and the condenser is frequently filled with steam of nearly boiler pressure owing to the leakage of valves and pistons when the engine is at rest with full steam pressure in the boilers. All that was required to produce a satisfactory "vacuum" was to clean the mud out of the condenser, and keep it cleaned out.

The high temperature (158°) of the feed water may be thought an economic compensation for the low "vacuum," but such is by no means the case. For example, suppose the temperature of the condenser to have been, say, 100°, and the corresponding

back pressure against the piston of the large cylinder to have been, exclusive of cushioning, say, 3.05 pounds per square inch. The experimental back pressure against this piston was, exclusive of cushioning, 6.05 pounds per square inch; the difference is 3 pounds per square inch to be added to the "equivalent" experimental indicated pressure, thus increasing the indicated horses-power developed by the engine $\left(\frac{3 \times 100}{30.17} =\right)$ 9.94 per centum. The number of units of heat imparted in the boilers to the pound of feed water at 158° to convert it into steam of the boiler pressure, is 1,059.53057; and the number that would have been required had the temperature of the feed water been 100°, is 1,118.00194; the gain by temperature being therefore $\left(\frac{1,118.00194 - 1,059.53057 \times 100}{1,059.53057} =\right)$ 5.52 per centum. Hence the loss by the poor "vacuum," notwithstanding what was recovered by the correspondingly higher temperature of the feed water, was (9.94 — 5.52 =) 4.42 per centum—the price paid for not keeping the engine in good order. The work of the air pump was increased by the poor vacuum, and the loss of fresh water from the condenser was also increased by the greater weight of steam discharged per stroke by the air pump. As this loss of fresh water had to be replaced in the boilers by additional feed water pumped in through the meter, the cost of the power in feed water consumed per hour was apparently proportionally increased. Strictly speaking, therefore, the cost of the power in weight of feed water *as measured*, is greater than the truth by the weight of steam discharged by the air pump into the atmosphere. The effect is exactly the same as though the boiler leaked the weight of water equal to the weight of steam discharged by the air pump. The measured feed water is consequently the weight to supply this leakage as well as to furnish the steam for the power.

The weight of fresh water thus lost by the air pump during the nine hours of the experiment, calculated from the useful space displacement of the pistons of the two air pumps in cubic feet during that time, and the weight of the cubic foot of steam of the

condenser pressure, deducting the space in cubic feet occupied by the feed water at the condenser temperature during that time, is 6,027.647459 pounds, or $\left(\frac{6{,}027.647459 \times 100}{595{,}620} =\right)$ 1.012 per centum of the feed water.

The weight of water in the boilers being 61.51 tons or 137,782.4 pounds, and the weight of steam discharged by the air pump being $\left(\frac{6{,}027.647459}{9} =\right)$ 669.7386 pounds per hour, there would be required $\left(\frac{595{,}620}{669.7386} =\right)$ 889⅛ hours to completely change the entire water in the boilers.

The water primed or foamed over from the boilers does not influence this question, as all such water in the case of surface condensers is received by the hot well and returned to the boilers by the feed pump.

There is a very probable source of water leakage from boilers in use, which may at times be large, caused by want of tightness in the "blow-off" valves communicating with the sea. Whether these valves leak or not can be easily ascertained by placing the hand on the "blow-off" pipe at as great a distance as possible outboard from the valve. If the valve be tight, the temperature of the pipe will be but little above that of the sea water; if the valve leaks, the temperature of the pipe will be nearly as high as that of the water in the boiler. This test is rarely made in practice. There should always be a tight cock between the boiler and its "blow-off valve," and this cock should be kept closed except when water is to be blown from the boiler. The ordinary disc valve cannot be depended on, and it may leak much without being suspected. Additionally, there is always a slight steam leakage from the safety valve—too small to attract notice—and considerable water is lost by the frequent trials of the gauge cocks. There may also be small unnoticed leakages from the inner seams and rivets, etc., of the boiler when under pressure. As a practical fact of almost universal occurrence, the loss of fresh water from boilers considered to be in good condition and well cared for is enormously greater than can be accounted for by the air-pump discharge of steam.

On the last two lines, under the head of "Engine," will be found the number of times the steam was expanded in the small cylinder and in the engine. Of course, these figures express the expansion of the steam only approximately; they express, in fact, only the ratio of the spaces occupied by the steam before and after the closing of the cut-off valve of the small cylinder. The steam, after the closing of that valve, undergoes liquefaction in part, and the resulting water vaporizes in part, so that its true measure of expansion cannot be known. Further, the steam which enters the small cylinder previous to the closing of the cut-off valve has undergone some expansion, but to an unknown amount, so that really it has done work by expansion from the beginning of the stroke of the piston of the small cylinder to the end of the stroke of the piston of the large cylinder.

The pressure in the boilers was 120,63 pounds per square inch above zero; in the valve chest of the small cylinder it had fallen 6.40 pounds per square inch to 114.63 pounds per square inch, although the length of the steam pipe could not have exceeded 25 feet, while the steam flow being stopped at five-eighths of the stroke of the piston of the small cylinder, had the time of the remaining three-eighths of the stroke to recover the pressure. From the steam chest of the small cylinder to the initial pressure in it was a further fall of 6.23 pounds of pressure, and between the commencement of the stroke of the piston of that cylinder and the point at which the steam was cut off there was another fall of pressure of 12 pounds per square inch. Thus, from the boiler to the point of cutting off the steam in the small cylinder the pressure fell continuously 24.63 pounds, or about one-fifth of the boiler pressure reckoned from zero; this corresponds to a considerable amount of expansion, during which the steam was performing molecular work upon itself and mass work upon external matter. The steam liquefaction attending the performance of these works, though unknown, must have been considerable, due to the transmutation of the heat of the steam into the works.

The ratio of the paddle surface to the size of the vessel was enormous, as was also the dip of the paddles, which was a trifle

over 50 per centum of the radius of the paddle wheel; nevertheless, the slip of the center of pressure of the paddle wheels may be considered excessive also, being 23.439 per centum of the circumferential speed of the center of pressure of the paddles. This slip is exactly the same that it would be with a common radial paddle wheel of the same diameter, the same length of paddles, and the same immersion, the paddles in both cases having the same aggregate surface.

Both types of paddle wheel, under the equal conditions just defined, are equally efficient propelling instruments, that is, they will have equal slips when driving the same vessel at the same speed, or overcoming the same resistance, for the mere fact of the paddles being vertical or inclined does not in the slightest degree affect their propelling efficiency; but the vertical paddle wheel is less *economically* efficient because the friction of the numerous journals upon which the paddles vibrate, the friction of the eccentrics producing these vibrating movements, and the friction of the connecting rods from the straps of the eccentrics to the levers of the paddles, must consume considerable power. And the increased weight of the wheel above what a common radial wheel of the same diameter would have weighed, gives the vessel that much more immersion which had to be propelled at the expense of engine power. The money cost of constructing the vertical paddle wheel is obviously greater than that of the radial paddle wheel by the expenditure required for the mechanism producing the verticalness of the paddles. In fact, this mechanism, being wholly useless, makes the vertical paddle wheel just that much commercially inferior to the radial paddle wheel.

Neglecting the comparatively small losses of useful effect due to immersing and withdrawing the bulk of the paddles, and due to overcoming the resistances of their wetted surfaces at the velocity of the immersion and withdrawal, and due to overcoming the resistance of the wetted surfaces of the arms and rings of the wheel caused by the velocity of the slip of the wheel, the losses of useful effect by either kind of paddle wheel when the lowest point of the rolling circle is at or above the surface of the

water, is due to the slip of the paddles alone and is proportional to it. But when the lowest point of the rolling circle descends below the water surface, there is the additional loss (and it is the same for both kinds of paddle wheel,) caused by the paddle surface lying between the horizontal lines of the water surface and of the tangent to the lowest point of the rolling circle, lifting a mass of water above the general water level, which mass then flows out over the surrounding water by gravity.

In a given paddle wheel with a given slip, and with the lowest point of the rolling circle below the water surface, any increase of slip will lessen the vertical distance between the above two horizontal lines, and *vice versa.* The cross-section of the mass of water thus pushed on both sides of the vessel will be measured by the vertical distance between the two horizontal lines and by twice the length of the paddle of one wheel.

In the case of the *Ville de Douvres*, the mean horizontal speed of the center of pressure of the vertical paddles, while passing through the space between the surface of the water and the horizontal tangent to the lowest point of the rolling circle, is 15.1 geographical miles per hour, which, subtracted from the speed of the vessel, 17.1 geographical miles per hour, leaves 2 geographical miles per hour as the speed with which the water lying between those horizontal lines acted against the paddle surface lying between the same lines to produce a retarding effect upon the vessel's speed.

As the lowest point of the rolling circle was 0.4122 foot below the upper edge of the paddle at its deepest immersion, there were utilized propulsively of the paddle even in that position only 3.9211 feet of its breadth of 4.3333 feet. The paddles were, consequently, about 5 inches too broad even for their most favorable position. The only effect of the curvature given to the paddles was to increase their resistance to immersion and emersion; it had no effect upon their propelling efficiency.

The use of a vertical paddle wheel is a result of ignorant engineering. When it was invented there was a supposed loss by the oblique action of the paddles upon the water, additional to their loss by slip. This supposition was quite erroneous, and

although many attempts to use the wheel have been made and are still being made from time to time, the truth was always shown by the practical performance, and the wheel was in all cases abandoned, radial paddles being substituted with increased economic performance. It will continue to show its economic inferiority as a propelling instrument comparably with the radial wheel as often as it may be tried. The *Ville de Douvres* furnishes the latest example.

The propelling action of the paddles of either kind of wheel consists in their *horizontal pressure* on the water in the opposite direction to the vessel's movement. The circumferential action of the paddles in the case of the radial paddle wheel is only an oblique application of the *power*, and, as the vessel cannot be moved by the pressure of the paddles in the vertical direction, the whole of the power is necessarily applied in the horizontal direction.

The horizontal reaction of the water against the paddle surface is in the ratio of the area of the projection of that surface upon a vertical plane, without regard as to whether the surface itself be vertical or inclined, consequently the same *projected* surface will, in either case, have the same propelling efficiency when moved with the same horizontal velocity, and if applied to the propulsion of the same vessel at the same speed, that is, overcoming the same resistance, will have exactly the same slip. If this same projected surface act at the end of a lever of the same length (the radius of the wheel) it will have, in both cases, the same horizontal velocity.

In all cases of mechanical action, the two forces involved, namely, those producing and those resisting motion, must be determined in value for the same direction. As the foregoing applications of mechanical philosophy, simple as they seem and indisputable as they are, have been completely overlooked by the builders of such paddle wheels as those of the *Ville de Douvres*, the pointing of them out cannot be considered superfluous.

Under the head of " Equivalent Pressures on the Piston of the Large Cylinder" (Table of Data and Results), will be found that the equivalent back pressure, exclusive of cushioning, against

this piston was 7.9811 pounds per square inch. This back pressure includes the pressure between the back pressure against the piston of the small cylinder, exclusive of cushioning, and the steam pressure on the piston of the large cylinder previous to the cutting off of the steam in the large cylinder, reduced in the ratio of the difference of the net areas of the pistons of the two cylinders. Now as the back pressure against the piston of the large cylinder, exclusive of cushioning, was 6.05 pounds per square inch, the pressure lost by transferring the steam from the small cylinder to the large one was (7.9811 − 6.05 =) 1.9311 pounds per square inch of the piston of the large cylinder, or (3.7929 × 1.9311 =) 7.3245 pounds per square inch of the piston of the small cylinder. Had there been no such loss, the equivalent net pressure (28.67 pounds per square inch) on the piston of the large cylinder would have been increased 1.9311 pounds per square inch, making the economy of the performance 6.7356 per centum greater.

The cause of this great loss of pressure represented by the "gap" between the portions described by the indicator diagrams from the two cylinders is the large "drop" in the small cylinder, the "drop" being the difference between the pressure of the steam in that cylinder at the end of the stroke of its piston and the back pressure against that piston, exclusive of the cushioning. The greater the "drop" the greater will be the "gap," and this cause of loss can only be avoided by making the small cylinder of greater capacity and expanding the steam more in it, so that, preserving the same indicated pressure in the small cylinder, the pressure at the end of the stroke of its piston will be very much less, the back pressure remaining the same. In fact, the "drop" should be reduced to nearly nothing for maximum economy.

The mechanical necessity of employing piston valves for steam of very high pressure, which makes the waste spaces at the ends of the cylinders to vary from one-tenth to one-fifth of the space displacement of their pistons per stroke, and the inconvenience of employing independent cut-off valves on the cylinders, substituting cutting off by steam lap on the valves with the result-

ing too much "release" and too little expansion, compelling, in consequence of the latter, the use of cylinders of too small capacity for the initial cylinders relatively to the following ones, thereby making the "drop" and the "gap" much too large, seriously impairs the economy otherwise derivable from the high-pressure steam used in multiple-cylinder engines. It is not the type of engine which gives the economy, but the regimen of the steam, and one type is preferable to another only as it practically allows a better regimen to be used.

The superiority of the Corliss valve gear used for land engines with moderately-sized cylinders, consisted in the simple fact of its reducing the waste spaces at the ends of the cylinders to about 3 per centum or less of the capacity of the cylinder, whereas the valve gears employed with the usual land engines had double that percentage or more. The greater measure of expansion employed with the Corliss valve gear reduced the "drop," and thereby lessened the mean back pressure against the piston, causing a corresponding economic gain. The steam was expanded more than was necessary for deriving all the benefit possible from the expansion *per se*, but the additional expansion increased the economy by diminishing the "drop," and consequently the mean back pressure, the vacuum in the condenser remaining constant.

Intelligent engineering arranges its mechanism for the production of the most economic regimen of the steam without regard to types of engine, selecting them only as necessary to practically enable that regimen to be used. If steam of the best regimen could be practically used in a single-cylinder engine, the resulting economy would be greater than could be obtained from its use in any type of multiple-cylinder engine. The latter is only accepted because the practical conditions cannot be obtained in the single-cylinder engine. Ignorant engineering ignores the regimen and believes the economy is in the type of engine, as a fetish, regardless of the regimen. It attempts to make light engines by sacrificing the economy of the steam, without considering that the boilers are thereby made heavier and the coal bunkers larger. A competent examination of the different regimens

with which the steam is used in different engines of the same type would explain most of the differences of economy with which the power is found to be developed in them.

Under the head of "Expanding Pressures Compared," are given, first, the mean pressure in pounds per square inch above zero of the expanding steam after the closing of the cut-off on the small cylinder, calculated according to the Mariotte law. This expansion extends, of course, from that point of cut-off through the waste space at the end of the large cylinder to the end of the stroke of the piston of that cylinder. This mean pressure is 33.757 pounds.

The experimental mean pressure of the expanding steam is next given as measured from the indicator diagrams *for the stroke of the pistons*, consequently, it will be less than the calculated expanding pressure by the pressure lost in the waste space between the piston of the small cylinder at the end of its stroke, and the piston of the large cylinder at the commencement of its stroke, and by the pressure lost between the points of releasing the steam and the end of the stroke of the pistons of the two cylinders. The mean pressure thus measured was 28.527 pounds. If proper allowance be made for the differences due to the causes just stated, there will be found that the two pressures are almost exactly the same. The most of this difference of 5.230 pounds per square inch, is due to the loss of pressure between the release of the steam (at 0.85 of the stroke of the pistons) and the end of the stroke of the pistons of the two cylinders, a large sacrifice of economy due to a poor arrangement of valve gear.

Experiments with many engines of different types and sizes, using steam with a great many measures of expansion, have shown that the mean pressure of the expanding steam alone, as measured on the indicator diagram, is almost exactly the same as the mean pressure calculated by the Mariotte law. Accepting this equality, and omitting the influence of the waste spaces at the ends of the two cylinders, the steam would be expanded 6.064 times, consequently, assuming that number to represent the whole stroke of the pistons, the space occupied by the unexpanded steam would be $\left(\frac{1.000}{6.064} = \right)$ 0.16 and the space occupied

by the expanded steam would be (1.00 — 0.16 =) 0.84; hence, if the loss of pressure due to the release alone be taken at, say, 4 pounds per square inch for the portion of the stroke occupied by the expanded steam, it would amount for the whole stroke to (0.84 × 4. =) 3.36 pounds per square inch, or to about $\left(\frac{3.36 \times 100}{28.67} = \right)$ 11.7 per centum of the net pressure or the pressure applied to the crankpins, a very serious deduction for the economy of the performance. The larger the "drop," other things equal, the larger will be the loss of useful effect due to the release. With a very small "drop," the loss by even a considerable percentage of release, is insignificant.

It is of the utmost importance in presenting the results of an experiment, that all the conditions be accurately described, that the dimensions and kind of mechanism be fully stated, and that every fact possible of measurement should be measured and given with the necessary calculations and deductions relating it to other facts, either general or special to the trial, whether the experimenter considers such facts necessary to his purpose or not. The results depend upon the limitations of the trial; they are relative, not absolute, and these limitations include the regimen of the steam.

The surrounding of the cylindrical portion of the small cylinder with its own exhaust steam, had no effect upon the economy of the performance; it was both beneficial and injurious, and the one just balanced the other, leaving the result the same as though no such jacketing had been applied.

The equivalent total pressure on the piston of the large cylinder (44.7738 pounds per square inch above zero), and the succeeding equivalent back pressure against that piston (7.9811 pounds per square inch above zero), are, of course, exclusive of the cushioning of the steam.

Under the head of "Weight of Steam accounted for by the Indicator," will be found the weights present at the point of cutting off the steam in the small cylinder; at the release of the steam from that cylinder; and at the release of the steam from the large cylinder. These quantities are calculated from the

pressures at those points, and do not include all the steam that the indicator should account for. They include only the steam present as such, but do not include the quantities of steam liquefied by the transmutation of a portion of its heat into the molecular and mass work performed by the expanding steam, nor do they include the liquefactions caused by the alternate cyclic changes of temperature of the metallic walls of the cylinders.

As the steam expands from the commencement of the stroke of the piston of the small cylinder to the point of cutting off the steam therein, mass and molecular work must have been done by it within that space in proportion to the expansion of that steam, but as the measure of this expansion is unknown, the corresponding liquefactions cannot be calculated. After the closing of the cut-off on the small cylinder, however, the mass work done by the expansion of the steam between that point and the end of the stroke of the piston can be calculated from the indicator diagram, and its equivalent liquefaction of steam can be ascertained. This has been done. The mass work done by the expanding steam from one end of the stroke of the piston of the large cylinder to the other, can be measured from the indicator diagram and its equivalent liquefaction of steam can be ascertained. This has been done. The weight of steam, therefore, determined by the indicator includes the steam present as such in the cylinders, and liquefied to produce the mass work done by the steam in expanding from the point of cut-off in the small cylinder to the end of the stroke of the piston of the large cylinder. None of the liquefactions due to the molecular work of the expanding steam is included. No liquefaction due to the mass work performed by the expansion of the steam in the small cylinder previous to the closing of the cut-off is included, and no liquefaction caused by the alternating temperatures of the walls of the cylinders is included, consequently, the weight of steam accounted for by the indicator should be largely less than the weight evaporated in the boiler. But on examining the "Difference between the Weight of Steam Evaporated in the Boilers and the Weight of Steam accounted for by the Indicator," there is found that this difference at the point of

cutting off the steam on the small cylinder is only 14.975598 per centum of the boiler weight, a large portion of which was undoubtedly due to the influence of the cylinder walls, while at the release of the steam from the small cylinder, the discrepancy is only 15.426815 per centum, or nearly the same, notwithstanding the considerable mass and molecular work done by the expansion between those points, and the consequent liquefaction, showing that much vaporization of water of liquefaction must have occurred between the point of cutting off the steam and the release of the steam from the cylinder. Again, there is found at the release of the steam from the large cylinder, that there remains only 7.661937 per centum of the boiler steam unaccounted for, and, accepting the experimental measurements as exact, this per centum must equal the aggregate liquefactions due to the mass and molecular work of the steam in the small cylinder previous to the cutting off, the molecular work of the steam expanding from the cutting off in the small cylinder to the release of the steam from the large cylinder, and the liquefactions caused by the heat interchange of the metal of the cylinders. A very large vaporization of water of liquefaction must have occurred in the large cylinder. These cylinder vaporizations are, of course, due to the contained heat of the water of liquefaction and to the temperature of the metal under the lessening pressures produced by the expansion of the steam.

It is quite possible that, in this particular case, the principal portion of the heat imparted by the steam to the metal of the cylinders may have been restored by the vaporization of the waters of liquefaction in the cylinders; if so, then the principal portion of the 7.661937 per centum is the measure of the molecular work done by the expanding steam from the commencement of the stroke of the piston of the small cylinder to the point of release from the large cylinder, and of the mass work done by the steam in its expansion from the commencement of the stroke of the piston of the small cylinder to the point of cutting off the steam.

Had the Committee of Research ascertained the weight of injection water used, and its temperatures before and after use, the

number of units of heat in it subtracted from the number of units of heat imparted in the boilers to the feed water, would have given the number of units of heat transmuted into the mass work done by the expanding steam in the cylinders, and into the molecular work done by the steam during its expansion from the boilers to the condenser. The heat in the injection water must equal the heat in the feed water, less the heat annihilated in the production of work—mass and molecular. If from this remainder there be deducted the mass work done by the expanding steam in the cylinders after the closing of the cut-off on the small cylinder, and if a reasonable estimate be made of the mass work done by the steam during its expansion in the small cylinder previous to the closing of the cut-off, and this also be deducted, the last remainder will be the number of units of heat transmuted into the molecular work of the steam from its departure from the boiler to its arrival into the condenser.

The molecular work of steam, which is that of overcoming the cohesion of its own molecules during its expansion, no doubt varies with its pressure; the higher the pressure, accompanied by the higher corresponding temperature, the less will be the cohesion of the molecules, and the less proportionally will be the work of overcoming that cohesion.

The well-known thermodynamic equation of Clausius for calculating the specific volume of vapors is deficient of a factor representing the force of cohesion of the vapor molecules in function of temperature; consequently the steam tables, which have been calculated according to this formula, are erroneous to that extent—they give too large a specific volume. This error has not before been pointed out, but it seriously vitiates the value of the tables. At higher temperatures the error is, of course, less than at lower ones, and, although direct experiments on the specific volume of steam cover but a small portion of practical pressures, they support, as far as they go, this deduction.

Under the head of "Economic Results" is given the cost of the indicated, net and total horses-power, in pounds of water, in Fahrenheit units of heat, in pounds of coal, and in pounds of combustible consumed per hour. The proper cost of the power

for comparison is its cost in units of heat consumed per hour. The total horses-power developed by the engine during a given time represent the entire mass work done by the steam produced by the water, heat or fuel consumed in that time. The net horses-power represent the commercially available work, or work done external to the machine. The difference between these two powers is the power expended in working the engine *per se*, and in overcoming the back pressure against its pistons, exclusive of the cushioned back pressure.

Under the head of "Vaporization" is given the economic results in pounds of water vaporized under the atmospheric pressure and from the temperatures 100 and 212 degrees Fahrenheit by one pound of the crude coal consumed, and by one pound of "combustible" consumed, or by one pound of what remains of that coal after deducting the ash. These quantities, in fact, are too small for the assumed conditions. They would be correct if the temperature of the water in the boiler had been 212 degrees, but as it was 341.44 degrees Fahrenheit, and as the temperature of the gases of combustion in the chimney is affected by the temperature of the water undergoing vaporization, the higher that temperature, the higher, necessarily, being the temperature of these gases, and the higher this latter temperature the less being the economic vaporization; also, as the higher the temperature of the boiler water the greater is the loss by external radiation of heat, there follows that the tabular economic vaporizations must be too small for the conditions stated, and in some ratio to the difference between the assumed temperature, 212 degrees, and the experimental temperature, 341.44 degrees, of the boiler water. The question is, Can the economic vaporization under one set of conditions be calculated for another set, or must the determination be purely experimental for every difference?

According to all the observations the writer has been able to make during his many experiments on boiler vaporization, the effect of increasing the temperature of the water in the boiler is to increase the temperature of the gases of combustion in the chimney by an equal amount; that is to say, an increase of, say, 100 degrees in the temperature of the boiler water is accompa-

nied by an increase of 100 degrees in the temperature of the chimney gases, other things, of course, remaining equal. The writer's observations have been limited to temperatures between 212 and 600 degrees Fahrenheit, but between them the *pari passu* increase has remained practically true. The difficulties of accurately obtaining the mean temperature of the chimney gases must, in this connection, not be overlooked. Evidently this equal rise of the temperature of the gases of combustion in the the chimney, with rise of the temperature of the boiler water, is exactly true for the extreme case of a boiler in which the area of the heating surface and the rate of combustion are so managed that the gases of combustion leave the boiler with the temperature of the contained water, and they cannot under any circumstances leave the boiler with a less temperature.

Now the effect of increasing the temperature of the chimney gases is to increase the quantity of heat lost in them as sensible heat and as heat transmuted into their work of displacing the atmosphere during their escape into it. These quantities of heat are calculable when the necessary data is had, and being calculated for the same weight of gases discharged at different temperatures in equal time, the loss of heat due to these temperatures can be known, and being known, the economic vaporization experimentally ascertained for one temperature can be calculated for any other temperature, by using the difference between the losses of heat due to the temperatures of the chimney gases in the two cases; that is, by finding out how much vaporization would be given by the difference of the units of heat lost in the two cases, and adding or subtracting it, as required, to the economic vaporization obtained in the experimental case. Applying this principle to the experimental economic vaporization of the boilers of the *Ville de Douvres*, their economic vaporization, had the temperaturee of the water in those boilers been 212 instead of 341.44 degrees Fahrenheit, will be obtained as follows:

The difference between the temperature of the boiler water in these two cases is 129.44 degrees, so that the temperature of the gases of combustion in the chimney would have been, with the

water in the boiler at 212 degrees, (950.00° — 129.44° =) 820.56 degrees. As the quantity of sensible heat lost in these gases per pound of combustible consumed was 4,128.521051 units, when the chimney gases had the experimental temperature 950 degrees, it will be 3,566 units when their temperature is 820.56 degrees.

The work done in displacing the atmosphere by the gases of combustion at the experimental temperature of 950 degrees, was equivalent to a heat transmutation of 1,084.734093 units per pound of combustible consumed.

The bulk of atmospheric air per pound of combustible entering the ashpits in both cases, will be the same, namely, 271.128479760898 cubic feet, but the bulk of the gases of combustion in the chimney which, at the temperature of 950 degrees (absolute temperature 1,409.53 degrees), was 675.916367-546122 cubic feet, will be at the temperature 820.56 degrees (absolute temperature 1,280.09 degrees), only 613.845596001586 cubic feet, the diminution being in the ratio of the two absolute temperatures. Consequently, the difference between the bulk of air entering the ashpits and the bulk of gases of combustion leaving the chimneys, will be for the 950 degrees temperature 404.787887785224 cubic feet, and for the 820.56 degrees temperature 342.717116240688 cubic feet, and as the work done will be in proportion to these two quantities, and as the heat transmuted is in proportion to the work done, the heat transmuted in the case of the 820.56 degrees temperature will be 918.399370 units.

The heat lost by radiation per pound of combustible consumed will be in the ratio of the difference of the temperature of the air in the fire rooms and of the water and steam in the boilers. In the case of the 950 degrees temperature of chimney gases, this difference was (341.44 — 100.00 =) 241.44 degrees; and in the case of the 820.56 degrees temperature of the chimney gases, it would be (212 — 100 =) 112 degrees; and as the quantity of heat radiated in the first case was 751.997118 units, it would be in the second case $\left(\frac{751.997118 \times 112}{241.44} =\right)$ 348.838955 units.

The quantity of heat lost by the vaporization in the furnaces under the atmospheric pressure and from the temperature of the air in the fire rooms, of the hygroscopic water in the pound of combustible, will, of course, be the same in both cases, namely, 27.450498 units. The heat non-utilized in the vaporization of water in the boilers, per pound of combustible consumed, will, therefore, be as follows for the cases of the two temperatures of the water in the boilers, namely, 341.44 and 212 degrees Fahrenheit, corresponding to the two temperatures 950 and 820.56 degrees.

	Units of heat non-utilized in the vaporization of water in the boilers when the temperature of that water is 341.44 degrees Fahrenheit.	Units of heat non-utilized in the vaporization of water in the boilers when the temperature of that water is 212 degrees Fahrenheit.
Heat in the gases of combustion leaving the boilers above the temperature of the air in the fire rooms	4,128.521051	3,566.000000
Heat transmuted into the work of displacing the atmosphere by the gases of combustion leaving the boilers	1,084.734093	918.399370
Heat lost by radiation from the surfaces of the boilers	751.997118	348.838955
Heat vaporizing the hygroscopic water of the "combustible"	27.450493	27.450493
Total heat non-utilized	5,992.702755	4,860.688818
Heat utilized	10,234.797245	11,366.811182
Total heat of combustion of one pound of combustible	16,227.500000	16,227.500000
Ratio of the heats utilized	1.000000	1.110604

The economic experimental vaporization must, therefore, be increased in the above ratio in order to obtain what it would have been had the water in the boiler been vaporized under the atmospheric pressure instead of under the experimental boiler pressure of 105.8 pounds per square inch above the atmosphere. The pound of combustible, according to the table of data and results, vaporized 10.598320 pounds of water from the feed-water

temperature of 212 degrees, the vaporization being actually performed under the experimental boiler pressure. Had it, however, been performed under the atmospheric pressure the pound of combustible would have vaporized ($10.598320 \times 1.110604 =$) 11.770537 pounds of water from the same temperature of feed.

From the above will be perceived that increasing the boiler pressure from that of the atmosphere to that of, say, 106 pounds per square inch above the atmosphere, all other things remaining the same, decreased the economic vaporization one-tenth.

When the economy of steam engine performance is estimated in pounds of water or in units of heat consumed per hour per horse-power developed, the boiler-water temperature as in question does not affect the result; in fact, this mode of estimating the economic development of power entirely ignores the boiler effects as a factor in the problem; but when the economy of engine performance is estimated in pounds of fuel consumed per hour per horse-power, then for the same fuel consumed at the same rate in the same boiler, the economy is greatly affected by the boiler-water temperature, in the manner stated, for, all other things being equal, less water is vaporized per pound of fuel, with higher boiler-water temperatures than with lower ones. The relative economies of different engines, and of the same engine with different regimens of the steam, will be very different, supposing the same boiler employed and consuming the same fuel at the same rate, but generating steam from different boiler water temperature, whether the cost of the horse power is estimated in pounds of water, in units of heat, or in pounds of fuel consumed per hour. When different boiler-water temperatures are used in comparative experiments, the results obtained in pounds of water or in units of heat consumed per hour per horse-power developed, should be modified for the effect of the difference of such temperature as above illustrated. When that is done the true commercial comparison is had.

TABLE CONTAINING THE DATA AND RESULTS OF THE EXPERIMENT MADE ON THE EIGHTH OF SEPTEMBER, 1891, BY THE COMMITTEE OF RESEARCH, WITH THE BELGIAN POSTAL PADDLE-WHEEL STEAMER, *VILLE DE DOUVRES*, PLYING BETWEEN THE CITIES OF OSTEND AND DOVER.

TOTAL QUANTITIES.

Duration of the experiment in consecutive hours	9.
Total distance run by the vessel in geographical miles	153.9
Total number of double strokes made by the pistons	19,885.
Total number of pounds of coal consumed	66,420.
Total number of pounds of refuse from the coal	4,760.
Total number of pounds of combustible consumed	61,660.
Per centum which the refuse is of the crude coal	7.1665
Total number of pounds of feed water consumed by the main engine	556,470.
Total number of pounds of feed water consumed by the auxiliary engine driving the circulating pump	18,630.
Total number of pounds of feed water consumed by the auxiliary engines driving the centrifugal blowers	20,520.
Aggregate number of pounds of feed water vaporized in the boilers	595,620.
Per centum of the aggregate feed water consumed by the main engine	93.427017
Per centum of the aggregate feed water consumed by the auxiliary engine driving the circulating pump	3.127833
Per centum of the aggregate feed water consumed by the auxiliary engines driving the centrifugal blowers	3.445150

ENGINES.

Number of double strokes made per minute by the pistons of the main engines	36.824074
Number of double strokes made per minute by the piston of the auxiliary engine driving the centrifugal pump delivering injection water	240.
Number of double strokes made per minute by the pistons of the auxiliary engines driving the centrifugal blowers	404.
Height of the barometer, in inches of mercury	30.21
Pressure of the atmosphere, in pounds per square inch	14.83
Vacuum in the condenser, in inches of mercury	20.60
Air pressure above atmosphere, in inches of water column, in fire room	1.07
Steam pressure in boilers, in pounds per square inch above atmosphere	105.80
Steam pressure in boilers, in pounds per square inch above zero	120.63
Steam pressure in the valve chest of the small cylinder, in pounds per square inch above zero	114.23

Steam pressure in receiver, in pounds per square inch above zero...	26.83
Back pressure in condenser, in pounds per square inch above zero..	4.72
Pounds of feed water consumed per hour by main engine alone...	61,830.
Number of Fahrenheit units of heat consumed per hour by the main engine alone, taking for this number the number between the quantity due to the temperature of the feed water and the quantity due to the initial pressure of the steam in the small cylinder, multiplied by the number of pounds of feed water consumed per hour by the main engine.....	65,368,265.0310
Fraction of the stroke of the piston of the small cylinder and of the large cylinder completed, when the steam was cut off.....	0.625
Fraction of the stroke of the piston of the small cylinder and of the large cylinder completed, when the steam was released.....	0.85
Fraction of the return stroke of the piston of the small cylinder and of the large cylinder completed, when the back pressure was cushioned.....	0.18
Number of times the steam was expanded in the small cylinder..	1.484
Number of times the steam was expanded in the engine.....	5.482

TEMPERATURES.

Temperature, in degrees Fahrenheit, of the steam in the boilers...	341.44
Temperature, in degrees Fahrenheit, of the air on deck.....	64.0
Temperature, in degrees Fahrenheit, of the injection water.....	61.7
Temperature, in degrees Fahrenheit, of the discharge water.....	85.0
Temperature, in degrees Fahrenheit, of the feed water.....	153.0

SPEED AND SLIP.

Speed of the vessel per hour in geographical miles.....	17.1
Diameter of the rolling circle in feet.....	14.9932
Deepest immersion of the rolling circle in feet.....	1.8300
Diameter in feet of the center of pressure of the paddles taken 0.375 of their breadth from their outer edge.....	19.5833
Slip of the center of pressure of the paddles in geographical miles per hour.....	5.23514
Slip of the center of pressure of the paddles in per centum of their speed.	23.439

RATE OF COMBUSTION.

Pounds of coal consumed per hour for main engine alone.....	6,894.913854
Pounds of combustible consumed per hour for main engine alone..	6,400.788743
Pounds of coal consumed per hour per square foot of grates.....	31.3376
Pounds of combustible consumed per hour per square foot of grates.....	29.0918
Pounds of coal consumed per hour per square foot of heating surface (inside of tubes).....	1.1261
Pounds of combustible consumed per hour per square foot of heating surface (inside of tubes)	1.0454

STEAM PRESSURES IN SMALL CYLINDER.

Pressure on piston of small cylinder at commencement of its stroke, in pounds per square inch above zero	108.00
Pressure on piston of small cylinder at point of cutting off the steam, in pounds per square inch above zero	96.00
Pressure on piston of small cylinder at release of the steam, in pounds per square inch above zero	70.00
Mean back pressure against piston of small cylinder during its stroke, in pounds per square inch above zero	34.76
Mean back pressure against piston of small cylinder, exclusive of cushioning, in pounds per square inch above zero	32.33
Back pressure against piston of small cylinder at the point where the cushioning begins, in pounds per square inch above zero	33.00
Indicated pressure on the piston of the small cylinder, in pounds per square inch	55.49
Net pressure on the piston of the small cylinder, in pounds per square inch	52.64
Total pressure on the piston of the small cylinder exclusive of cushioning, in pounds per square inch above zero, the back pressure being reduced in the ratio of the length of the stroke of the piston to the fraction completed at the stroke when the steam was cushioned	82.00
Mean pressure of the expanding steam alone in the small cylinder after the closing of the cut-off valve, in pounds per square inch above zero	67.33
Mean pressure of the admission alone in the small cylinder previous to the closing of the cut-off valve, in pounds per square inch above zero	104.00

STEAM PRESSURES IN LARGE CYLINDER.

Pressure on piston of large cylinder at commencement of its stroke, in pounds per square inch above zero	29.83
Pressure on piston of large cylinder at point of cutting off the steam, in pounds per square inch above zero	21.83
Pressure on piston of large cylinder at release of the steam, in pounds per square inch above zero	17.00
Mean back pressure against piston of large cylinder during its stroke, in pounds per square inch above zero	6.33
Mean back pressure against piston of large cylinder, exclusive of cushioning, in pounds per square inch above zero	6.05
Back pressure against piston of large cylinder at the point where the cushioning begins, in pounds per square inch above zero	6.00
Indicated pressure on the piston of the large cylinder, in pounds per square inch	15.54
Net pressure on the piston of the large cylinder, in pounds per square inch	14.79

Total pressure on the piston of the large cylinder, exclusive of cushioning, in pounds per square inch above zero, the back pressure being reduced in the ratio of the length of the stroke of the piston to the fraction completed of the stroke when the steam was cushioned	20.50
Mean pressure, in pounds per square inch above zero, of the expanding steam on the piston of the large cylinder, including cushioning	21.87

EQUIVALENT PRESSURES ON THE PISTON OF THE LARGE CYLINDER.

Indicated pressure, in pounds per square inch, that would be on the piston of the large cylinder were the indicated pressure on the piston of the small cylinder divided by the ratio of the net areas of those pistons and the quotient added to the experimental indicated pressure on the piston of the large cylinder	30.17
Net pressure, in pounds per square inch, that would be on the piston of the large cylinder were the net pressure on the piston of the small cylinder divided by the ratio of the net areas of those pistons and the quotient added to the experimental net pressure on the piston of the large cylinder	28.67
Total pressure, in pounds per square inch above zero, that would be on the piston of the large cylinder, corresponding to the total aggregate horses-power developed by the engine	44.7738
Back pressure, in pounds per square inch above zero, that would be against the piston of the large cylinder, corresponding to its experimental back pressure plus the difference in pounds per square inch between the back pressure exclusive of cushioning against the piston of the small cylinder and the mean steam pressure on the piston of the large cylinder from the commencement of its stroke to the point of cutting off the steam, divided by the ratio of the area of the piston of the large cylinder to the area of the piston of the small cylinder	7.9811

EXPANDING PRESSURES COMPARED.

Mean pressure of the expanding steam, in pounds per square inch above zero, calculated according to the Mariotte law	33.757
Experimental mean pressure of the expanding steam, in pounds per square inch above zero	28.527

HORSES-POWER.

Indicated horses-power developed in the small cylinder	1,446.019512
Net horses-power developed in the small cylinder	1,371.751075
Total horses-power developed in the small cylinder	2,136.846279
Horses-power developed by the expanding steam alone in the small cylinder	657.992299
Indicated horses-power developed in the large cylinder	1,535.963932
Net horses-power developed in the large cylinder	1,461.834398

Total horses-power developed in the large cylinder	1,491.995676
Horses-power developed by the expanding steam in the large cylinder	1,591.704656
Aggregate indicated horses-power developed by the engine	2,981.983444
Aggregate net horses-power developed by the engine	2,833.585473
Aggregate total horses-power developed by the engine	3,628.841955
Aggregate horses-power developed by the engine by the expanding steam alone	2,249.696955

WEIGHT OF STEAM ACCOUNTED FOR BY THE INDICATOR.

Number of pounds of steam present per hour in the small cylinder at the point of cutting off the steam, calculated from the pressure there	52,570.587543
Number of pounds of steam present per hour in the small cylinder when the steam was released, calculated from the pressure at the point of release	50,462.971976
Number of pounds of steam liquefied per hour in the small cylinder to furnish the heat transmuted into the total horses-power developed by the expanding steam alone in that cylinder	1,828.628033
Sum of the two immediately preceding quantities	52,291.600009
Number of pounds of steam present per hour in the large cylinder when the steam was released, calculated from the pressure at the point of release	51,064.548941
Number of pounds of steam liquefied per hour in the small cylinder and in the large cylinder to furnish the heat transmuted into the total horses-power developed in those cylinders by the expanded steam alone after the closing of the cut-off on the small cylinder	6,028.075247
Sum of the two immediately preceding quantities	57,092.624188

DIFFERENCE BETWEEN THE WEIGHT OF STEAM EVAPORATED IN THE BOILERS AND THE WEIGHT OF STEAM ACCOUNTED FOR BY THE INDICATOR.

Difference, in pounds per hour, between the weight of steam evaporated in the boilers and the weight of steam accounted for by the indicator at the point of cutting off the steam in the small cylinder	9,259.412457
Difference, in per centum of the weight of steam evaporated in the boilers, between that weight and the weight of steam accounted for by the indicator at the point of cutting off the steam in the small cylinder	14.975598
Difference, in pounds per hour, between the weight of steam evaporated in the boilers and the weight of steam accounted for by the indicator at the point of release of the steam from the small cylinder	9,538.399991
Difference, in per centum of the weight of steam evaporated in the boilers, between that weight and the weight of steam ac-	

counted for by the indicator at the point of release of the steam from the small cylinder	15.426815
Difference, in pounds per hour, between the weight of steam evaporated in the boilers and the weight of steam accounted for by the indicator at the point of release of the steam from the large cylinder	4,737.375812
Difference, in per centum of the weight of steam evaporated in the boilers, between that weight and the weight of steam accounted for by the indicator at the point of release of the steam from the large cylinder	7.661937

ECONOMIC RESULTS.

Number of pounds weight of steam consumed per hour per indicated horse-power	20.734521
Number of pounds weight of steam consumed per hour per net horse-power	21.820411
Number of pounds weight of steam consumed per hour per total horse-power	17.038493
Number of Fahrenheit units of heat consumed per hour per indicated horse-power	21,921.0691
Number of Fahrenheit units of heat consumed per hour per net horse-power	23,069.0994
Number of Fahrenheit units of heat consumed per hour per total horse-power	18,013.5332
Number of pounds of coal consumed per hour per indicated horse-power	2.312190
Number of pounds of coal consumed per hour per net horse-power	2.433282
Number of pounds of coal consumed per hour per total horse-power	1.900031
Number of pounds of combustible consumed per hour per indicated horse-power	2.146487
Number of pounds of combustible consumed per hour per net horse.power	2.258909
Number of pounds of combustible consumed per hour per total horse-power	1.763875

VAPORIZATION.

Number of pounds of water that would have been vaporized in the boilers had it been supplied at the temperature of 100 degrees Fahrenheit and vaporized under the standard atmospheric pressure	585,125.851124
Number of pounds of water that would have been vaporized in the boilers had it been supplied at the temperature of 212 degrees Fahrenheit and vaporized under the standard atmospheric pressure	653.492.386977

Number of pounds of water vaporized under the atmospheric pressure from the temperature of 100 degrees by one pound of coal	8.809483
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 100 degrees by one pound of combustible	9.489553
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 212 degrees by one pound of coal	9.838790
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 212 degrees by one pound of combustible	10.598320

NOTE TO THE PRECEDING.

The French Naval Marine, about the year 1836, contracted with Mr. Cavé, at that time a French builder of machinery of distinguished reputation, for duplicate engines and boilers for the duplicate paddle-wheel war steamers *Tartare* and *Etna*. The hulls, the engines and the boilers of both vessels were exactly alike, but the paddle wheels were different. The *Tartare* was fitted with the ordinary radial paddle wheel, and the *Etna* was fitted with Cavé's vertical paddle wheel, in which the paddles entered, passed through, and left the water nearly vertical, being maintained during that time in this position by the action of an eccentric secured on the side of the vessel, with jointed connecting rods from the eccentric strap to the fixed arms on the paddles at right angles to the paddles. The long point of the eccentric was placed horizontally, and the position of the eccentric was such that the paddle-wheel shaft passed through it. The eccentric strap was carried around on the eccentric during a revolution of the wheel, by a fixed or rigid rod extending from the strap to a rim of the wheel and passing its free end between guide rollers secured to the latter.

The engines were of the side-lever kind in general use at that time, and were worked with steam of about 1.7 pounds per square inch above the atmosphere.

The vessels were of wood and copper sheathed, and as their speeds were always within the limit at which "squatting" occurs, their resistances were always in the ratio of the squares of their speeds. The purpose of fitting the two duplicate vessels with

paddle wheels of different types was to ascertain, by direct trial, their comparative efficiencies. That such a trial was deemed necessary is a singular proof of how little the principles of paddle-wheel propulsion were then understood by even the most scientific practitioners of marine engineering.

The following are the draughts of water of the vessels during the trials, and the principal dimensions of the paddle wheels:

	Tartare. Radial paddles.	*Etna.* Vertical paddles.
Vessel's draught of water, in feet, Forward	11.15506	11.13866
Vessel's draught of water, in feet, Mean	11.22068	11.15506
Vessel's draught of water, in feet, Aft	11.28630	11.17146
Diameter of paddle wheels to outside of paddles, in feet,	19.32450	20.07911
Number of paddles in each wheel	16.	8.
Length of each paddle, in feet	8.20225	6.56180
Breadth of each paddle, in inches	26.00000	28.34697
Area of each paddle, in square feet	17.77154	15.50060
Maximum immersion of the outer edge of the paddles, in feet	5.21790	5.42333
Maximum immersion of the inner edge of the paddles, in feet	3.05124	3.06108
Distance from the center of the wheel to the axis of vibration of the paddles of the *Etna*, in feet	.....	8.85843

When the vessels were completed, a number of experiments were made with them, running side by side, off the port of Cherbourg, in smooth water and calms, by a commission of French naval officers, one of whom was the celebrated naval engineer, Reech, from whose report the writer has taken the observed data, but nothing else.

In making the experiments, the absolute powers developed by the engines were not ascertained by the indicator, but they were ascertained relatively by assuming that in the case of identical engines, run at very nearly the same speed of piston, with the same boiler pressure, condenser vacuum, and cylinder valve arrangement, the throttle being wide open, the powers developed would be in the ratio of the piston speeds, an assumption undoubtedly true.

The mode of conducting the trials was, therefore, to run the vessels side by side at exactly equal speeds during about two hours, but sufficiently separated to prevent them from exerting

any effect upon each other, and ascertaining the number of revolutions of the wheels required per minute in each case, to produce this equal speed. Next, to run the vessels at the same time, in the same water and in the same direction, with equal boiler pressures and conditions above stated, measuring the relative speeds of the vessels by the product of the ratio of the number of revolutions made by the wheels per minute, multiplied by the ratio of the number of revolutions made by the wheels per minute during the trial with equal speeds of vessels.

As the data, according to this method of experimenting, could be obtained with great exactness, the relative results can be depended on as correspondingly near the precise truth.

Incidentally, the absolute speed of the vessels was obtained during their trial at equal speeds.

Two sets of experiments, one on the 27th and the other on the 31st of January, 1838, were made with the vessels at equal speed; they closely agreed, and the mean of the two is taken. Three sets of experiments, one on the 27th and the other two on the 31st of January, 1838, were made, with the vessels propelled by equal boiler pressures; they, also, closely agreed, and the mean of the three is taken. The following are the results:

TRIAL WITH THE VESSELS RUNNING SIDE BY SIDE AT EQUAL SPEED.

	Tartare. Radial paddles.	*Etna.* Vertical paddles.
Speed of the vessels per hour, in geographical miles of 6,086 feet	8.25	8.25
Number of revolutions made per minute by the paddle wheels	17.32	18.94
Ratio of the number of revolutions made per minute by the paddle wheels	1.000000	1.093533
Diameter, in feet, of the center of pressure of the paddles taken at 0.4 of their breadth from their outer edge	17.59116	18.18931
Speed of the center of pressure of the paddles, in geographical miles per hour	9.436535	10.670048
Slip of the center of pressure of the paddles, in geographical miles per hour	1.186535	2.420048
Slip of the center of pressure of the paddles, in per centum of its speed	12.573842	22.680760
Ratio of the speeds of the centers of pressure of the paddles	1.000000	1.130713

Ratio of the powers applied by the paddles to the water. This is exclusive of the power required to overcome the friction of the vibrating journals of the vertical paddles; consequently the ratio of the powers developed by the engines of the two vessels will be higher for the vertical paddle-wheel vessel by the unknown proportion that the above friction power is of the power applied to the water	1.000000	1.130713
Radius, in feet, of the rolling circle	7.69	7.03
Vertical distance, in feet, from the center of the paddle wheel to the water surface	4.44435	4.61622
Greatest immersion, in feet, of the rolling circle	3.24565	2.41378

The ratio (1.000000 to 1.130713) of the above powers is the ratio of the powers that would have been developed by the engines in the case of the respective vessels had there been no friction of the various journals of the vertical paddle-wheel vessel. Whatever that friction may have been, and it must have been considerable, as these journals vibrated with only water lubrication under the resistance of the vessel at its experimental speed, must be added to the power in the case of the vertical paddle wheel. Suppose this friction to have been $7\frac{1}{2}$ per centum, then the powers will compare as 1.000000 for the radial paddle wheel and (1.130713 × 1.075 =) 1.215516 for the vertical paddle wheel; that is to say, there will be required $21\frac{1}{2}$ per centum more power with a vertical paddle wheel to drive the vessel at a given speed than with a radial paddle wheel, the two having the proportions of the wheels experimented with. By increasing sufficiently the aggregate area of the vertical paddles to make their slip the same as that of the radial paddles, the difference in the economic effects would be exactly what was due to the friction of the journals of the vertical paddles.

As the vessels were propelled by the pressure of the paddles on the water, and as the vessels had the same speed, offering thereby the same resistance to be overcome, this pressure was the same in both cases; consequently, the powers developed *on the water* will be in the ratio of the speeds of the centers of pressure of the paddles (1.000000 and 1.130713). This ratio, however, is not the ratio of the powers developed *in the respective cylinders*, by the proportional power required to overcome the friction of the journals of the vertical paddle wheel.

The vertical paddles were both too few and too short to obtain the reaction from the water obtained by the radial paddles. They were one-half the number and four-fifths the length of the radial paddles, and it is amazing that any person could have imagined their greatly less surface would be propulsively as efficient. The aggregate surfaces in the two cases compare as 2.5 for the radial and 1.0 for the vertical paddles. Now, had the vertical paddle surface been increased in this proportion, it would have had a slip in the inverse ratio of the square roots of 2.5 and 1.0, that is of 1.581139 to 1.000000. The slip of the vertical paddles was 2.420048 geographical miles per hour, which divided by 1.581139 gives the quotient 1.5305726, and as the depth of the watery stream acted on by the paddles of the two wheels compares as (5.21790 — 3.24565 =) 1.97225 and (5.42333 — 2.41378 =) 3.00955, the square roots of which numbers compare as 1.40437 and 1.73480, or as 1.00000 and 1.23529, the division of the latter quantity into the 1.5305726, produces 1.239040 for the speed of what would be the slip in geographical miles per hour of the vertical paddles if they had the surface of the radial paddles. Adding this quantity to the speed of the vessel (8.25 geographical miles per hour) there is obtained 9.48904 geographical miles per hour as the speed of the center of pressure of the vertical paddles, which divided into the 1.239040 multiplied by 100, gives the slip 13.057610 per centum. The slip of the radial paddles was 12.573842 per centum, a very close approximation, showing what indeed required no experimental confirmation, that the propelling efficiency of the two types of paddle wheels, when they had the same dimensions and acted under the same conditions, was the same, the vertical paddles having always the economic inferiority due to the friction of their vibrating journals and eccentric strap.

There will be observed that the outer edges of the radial paddles in these comparative experiments had the excessive maximum immersion of 54 per centum of the radius of the wheel.

Notwithstanding these and other practical results of the same kind, Mr. Cavé continued, and after him others down to the

present time continue, to use the vertical paddle wheel, with a serene indifference to scientific reasonings and experimental facts.

TRIAL OF THE VESSELS WITH EQUAL BOILER PRESSURES.

(Powers developed in the cylinders in the ratio of the revolutions made in equal time by the respective paddle wheels.)

	Tartare, Radial paddles.	*Etna*, Vertical paddles.
Ratio of the speeds of the vessels	1.000000	0.928505
Ratio of the revolutions of the paddle wheels made in equal time	1.000000	1.015351
Ratio of the powers developed in the cylinders	1.000000	1.015351
Cube of the ratio of the speeds of the vessels	1.000000	0.800484
Quotient of the cube of the ratio of the speeds of the vessel divided by the ratio of the powers developed in the cylinders, expressing the relative economic efficiencies of the paddle wheels	1.000000	0.788382

The ratio of the powers in the above table is the ratio of the indicated horses-power.

The economic inferiority of the vertical paddle wheel is (1.000000 — 0.788382 =) 0.211618, the economic performance of the radial paddle wheel being taken as unity; or, the vertical paddle wheel is, roundly, $21\frac{1}{6}$ per centum less economically efficient than the radial paddle wheel, almost exactly the same result as that previously obtained by assuming the friction of the journals of the vertical paddles at Morin's coefficient of $7\frac{1}{2}$ per centum.

The conclusion of the French commission was strongly against the vertical paddle wheel. They reported the radial paddle wheel as $23\frac{1}{2}$ per centum more economically efficient than the vertical paddle wheel, which is sensibly the writer's result when the economic performance of the vertical paddle wheel is taken as unity.

The commission proposed to make a very interesting experiment that would have been decisive in determining the relative propelling values of the two types of wheel when identical paddles were employed. Permission was asked to disconnect the eccentric movement of the vertical paddle wheel and secure its paddles radially, but it could not be obtained.

The whole question at issue being simply the determination of the relative propelling efficiencies of the same paddle surface when in the radial and when in the vertical position, the French Admiralty might have saved itself the expense of constructing the additional vessel to answer it. One would, in fact, have been better than two. If the two arrangements of the same paddle surface gave, on trial, the same slip, as they certainly would have done, then the economic inferiority of the vertical paddles would have been shown to be solely due to the power absorbed by the friction of their journals.

A trial with equal boiler pressure, under these conditions, would have given equal per centums of slip, and a superiority of speed for the radial arrangement of paddles in the ratio of the cube roots of unity and of what remained of unity after deducting the ratio of the power required to overcome the friction of the journals of the vertical paddles. Such an experiment would, therefore, have determined the proportion of the engine power that was absorbed in this friction.

XXII.

SPEED TRIALS.

By David W. Taylor, Naval Constructor, U. S. N.

As we all know, in progressive speed trials simultaneous values of the speed of the ship, the indicated horse power, and the revolutions per minute are experimentally determined for a certain number of speeds, "progressing" from high speeds to low, or *vice versa*. By plotting the results of the experiments and striking fair curves through the experimental "spots" we obtain curves giving the relations between speed, power and revolutions within the extreme speeds covered by the trials.

The curves can be extended by eye slightly above the top speed, and if the lowest speed is not over five or six knots they can be extended to the origin.

Table I gives the essential data of the progressive trials of the *Yorktown* made in Narragansett Bay, near Newport, September 21, 1889. Fig. 1 shows curves of revolutions and indicated horse-power plotted from the results of Table I upon speeds as *abscissæ*.

Consider now for a moment the three variables, speed, I.H.P. and revolutions. It appears to my mind more natural and logical to treat the speed and the I.H.P. as "functions" of the revolutions, making the latter the independent variable, so to speak. The relation between revolutions and I.H.P. is more simple and direct than that between speed and I.H.P. For these and other reasons I prefer, in dealing with the results of trials, to plot curves of speed and I.H.P. upon revolutions as *abscissæ*.

Fig. 2 shows the same data used in Fig. 1 plotted in this way. It may be noted that the curve of speed plotted on revolutions is simply the curve of revolutions plotted on speed and revolved 180°, about a 45-degree line through the origin.

Table I.

DATA FROM TRIAL OF U. S. S. *YORKTOWN* ON THE MEASURED MILE IN NARRAGANSETT BAY, SEPTEMBER 21, 1889.

No. of run.	Time of taking card.	Starboard Engine.		Port Engine.		Average revolutions, port and starboard	Total I.H.P., both engines.	Apparent speed.
		Rev. per minute.	I.H.P.	Rev. per minute.	I.H.P.			
	h. m.							*Knots.*
1	8·41	36.95	32.9	39.02	33.5	37.98	66.4	3.47
2	9·15	42.39	39.9	38.47	35.6	40.43	75.5	5.08
3	9·50	39.19	32.3	38.80	39.4	39.00	71.7	3.91
4	10·24	39.05	35.8	41.73	51.8	40.39	87.6	4 77
5	2·08	91.49	298.0	98.24	373.8	94.87	671.8	10.57
6	2·21	94.30	267.0	97.01	346.9	95.66	613.9	10.50
7	2·37	98.03	369.7	97.12	322.6	97.58	692.3	10.77
8	2·51	92.81	300.0	98.65	366.4	95.73	666.4	10.56
9	11·06	136.17	1023.3	132.31	963.3	134.24	1986.6	14.53
10	11·18	136.20	984.7	137.88	1058.2	137.04	2042.9	14.75
11	11·30	144.35	1097.2	143.90	1159.1	144.13	2256.3	14.75
12	11·43	146.38	1231.6	147.65	1350 5	147.02	2582.1	15.46
13	12·01	160.95	1737.9	160.60	1731.7	160.78	3469.6	16.45
14	12·12	161.29	1734.8	161.40	1706.4	161.35	3441.2	16.95
15	12·22	160.54	1726.8	160.38	1710.1	160.46	3436 9	16.30
16	12·33	160.22	1681 7	161.00	1633.6	160.61	3315.3	16.97

	Average revolutions.	Average total I.H.P.	Final mean speed.
For first series, runs 1-4	39.45	75·3	4.40
" second series, runs 5-8	95.96	661.1	10.62
" third series, runs 9-12	140.61	2217.0	14.81
" fourth series, runs 13-16	160.80	3415.7	16.65

Actual speed trials, unfortunately, have to be made over courses exposed to a variable current. But let us suppose that we have a course in still water. A single run in either direction over such a course with accurate observations of speed, revolutions and horse power, would give data for one spot on our curves.

Assuming that the curves of Fig. 2 are exact, suppose we made 12 runs with the *Yorktown* over a still-water course, observing the speed and revolutions. We should, of course, obtain the same curve of true speed as before. Fig. 3 shows this curve of true speed, the + marks indicating the spots corresponding to 12 runs.

Suppose, as a next step, that our course is not in still water, but in a steady current. In this case, from observations on a run, we could obtain only the apparent speed of the vessel, *i. e.*, the true speed plus or minus the speed of the current. What is to be done to eliminate the effect of the current and obtain the curve of true speed?

1. A run with the current at a given number of revolutions gives the true speed plus the speed of current. If we make the next run at the same number of revolutions, but against the current, we will obtain the true speed minus the speed of current. The average of the apparent speeds of the two runs will give the true speed corresponding to the revolutions. Thus, from 12 runs, 6 spots for a speed curve can be obtained.

2. Suppose we make the 12 runs back and forth, just as on the still-water course, increasing the revolutions each time. We should thus obtain 6 spots for each of two curves, one representing speed of ship with the current, the other speed of ship against the current.

These curves are shown in Fig. 3, the data being still from the *Yorktown* speed curve, and the strength of the current being assumed $1\frac{1}{2}$ knots.

The curve of true speed lies half way between the curves of speed with and against current.

For accurate determination of speed alone there would be little choice between the two methods outlined above. The second would probably have a shade the best of it in practice, as it would be slightly easier to carry out.

In the case of a course where the current is variable, the mean of successive runs with and against the current will not, of course, give the true speed. From trials on such courses we must eliminate in some way the effect of the current.

The regulation method of accomplishing this result (see G. O., No. 375, issued by U. S. Navy Department,) is to make four successive runs—two with and two against the tide—and adopt as the true speed the final mean of the four apparent speeds.

Thus, for the *Yorktown*, at the ten-knot series (Table I), we have—

Apparent Speeds.	First Means.	Second Means.	Final Mean.
10.57	10.535	10.585	10.6175
10.50	10.635	10.650	
10 77	10.665		
10 56			

The *modus operandi* of the method is obvious. The reason for its adoption will appear below. It should be remarked that it is equivalent to giving to the first and last runs the relative weight of one, and to the two middle runs the relative weight of three. Or, if V denote the final mean speed, and V_1, V_2, V_3, V_4, the successive apparent speeds,

$$V = \tfrac{1}{8}(V_1 + 3V_2 + 3V_3 + V_4).$$

Applying this formula to the case of the *Yorktown* above, we have

$$V = \tfrac{1}{8}(10.57 + 3 \times 10.50 + 3 \times 10.77 + 10.56) = \tfrac{1}{8} \times 84.94 = 10.6175.$$

I shall consider now the principal objections to the regulation method.

1. It is very difficult, especially with twin-screw ships, to secure the same revolutions for four successive runs. Table I shows this clearly, yet the care and skill employed on the *Yorktown* trials were decidedly above the average.

2. Supposing uniform revolutions obtained, the effect of the tidal current is exactly eliminated by the adoption of the final mean only by accident. This fact is capable of rigid demonstration.

Let C denote the speed of current. It is a function of the time and hence of the number of the run. Let n denote the number of the run. Then take $C = a + bn + cn^2 + dn^3$.

The right-hand expression involves four arbitrary constants, a, b, c, d, and, since there are only four speeds of current, it is always possible to give a, b, c and d such values that the equation holds.

Let V denote the true speed, C_1, C_2, C_3, C_4, the successive speeds of current corresponding to n (the number of the run) = 1, 2, 3 and 4. Then

$$C_1 = a + b + c + d,$$
$$C_2 = a + 2b + 4c + 8d,$$
$$C_3 = a + 3b + 9c + 27d,$$
$$C_4 = a + 4b + 16c + 64d,$$
$$V_1 = V + C_1 = V + a + b + c + d,$$
$$V_2 = V - C_2 = V - a - 2b - 4c - 8d,$$
$$V_3 = V + C_3 = V + a + 3b + 9c + 27d,$$
$$V_4 = V - C_4 = V - a - 4b - 16c - 64d.$$

$$\begin{aligned}\text{Final Mean} &= V + \tfrac{1}{8}(a - 3a + 3a - a) \\ &\quad + \tfrac{1}{8}(b - 6b + 9b - 4b) \\ &\quad + \tfrac{1}{8}(c - 12c + 27c - 16c) \\ &\quad + \tfrac{1}{8}(d - 24d + 81d - 64d) \\ &= V - \frac{3d}{4}.\end{aligned}$$

Unless, then, the current changes during the four runs so that in the formula $C = a + bn + cn^2 + dn^3$ the co-efficient $d = o$, the final mean is in error. Now, while d is probably small, it will always have a value for a tidal current except by accident. There appears to be no easy way to determine its value in a given practical case. By making imaginary runs with all conditions such as might occur in practice, we can get at some idea of the intrinsic error involved in the final mean method, using four runs.

3. A serious objection to the final mean method is the large number of runs required to cover the ground satisfactorily. Sixteen runs give but four spots for our curves, and for purposes of critical analysis a curve through four experimental (and hence approximate) spots leaves much to be desired.

I have shown that, in the case of the uniform current course, accurate results were attainable by plotting the curves of apparent speed (with and against current) and using a mean curve as the true curve of speed.

The question naturally arises whether the same method will not give satisfactory results if used in a variable current.

Theoretically it will. The curve of speed plotted on revolutions is a fair curve. The curve of current plotted on time is a

fair curve, except about the turn of the tide, when speed trials should never be made. If about the same course be steamed over always, the intervals between successive runs in the same direction will, if plotted, form a fair curve.

The above are the necessary and sufficient conditions to render the curves of apparent speeds fair curves, and their mean the correct curve of true speed.

I regret that I cannot lay before the Society results of a progressive trial conducted as above. It is possible, however, to form a fair idea of the accuracy of the method I propose without actual trials. With a curve of speed for an actual ship and a curve showing tidal current at any time on a given course, we can tell what the apparent speed would be for a run over the course; and by making reasonable allowances as to intervals between runs and the revolutions of successive runs we can get a fair idea of what trial results would be if there were no errors in recording times, &c.

Figure 4 is a curve of strength of tidal current plotted on time. It is plotted from the results of tidal observations made by the Coast Survey in Long Island Sound in 1890.

The curve of speed in Fig. 2 is taken as representing an exact speed curve.

Now for theoretical exactness, using the method I propose, the same course should be steamed over—off the mile as well as on the mile—for each run. Also successive runs should be always made with a constant difference of revolutions. It is necessary to made allowances for unavoidable errors on each of these points. But first suppose no errors made. Let the fastest run be made at 160 revolutions, and suppose the revolutions gradually decreased by 9 each time (the same course being traversed always) until we reach 61. The revolutions on each run and the intervals between runs will be as in Table II below, starting with a reasonable minimum interval of 12 minutes. By interval between runs, I mean the time elapsed from the instant a vessel is at the middle of the mile, going one way, to the instant when she is next at the middle of the mile, going the other way.

Table II.

Number of Run.	Revolutions.	Interval in Minutes from Preceding Run.
1	160	—
2	151	12
3	142	13
4	133	14
5	124	15
6	115	17
7	106	18
8	97	20
9	88	22
10	79	24
11	70	26
12	61	28

Now, I have assumed various errors in the revolutions aimed at and in the intervals, and distributed them by lot to the 12 runs.

Table III.

Number of Run.	Revolutions.		Interval.		Time of middle of Run.	
	Desired	Obtained.	Desired.	Obtained.		
					h.	*m.*
1	160	161	...	...	9	11
2	151	154	12	13	9	24
3	142	141	13	12	9	36
4	133	134	14	17	9	53
5	124	123	15	14	10	7
6	115	117	17	19	10	26
7	106	104	18	18	10	44
8	97	97	20	18	11	2
9	88	87	22	23	11	25
10	79	81	24	21	11	46
11	70	71	26	28	12	14
12	61	61	29	29	12	43

The result is shown in Table III, where the actual revolutions supposed to be obtained range from 3 above to 2 below the number aimed at, and the intervals are from 3 minutes too great

to 3 minutes too small. Round numbers are used for convenience. I think it will be conceded that the columns of revolutions and intervals "obtained" are fair examples of what might be obtained in practice. These once settled, from the tidal current curve and the curve of speed, Table IV is readily filled in.

Table IV.

No. of run.	Time at middle of run.		Revolutions.	True speed.	Speed of tidal current.	Apparent speed.
	h.	*m.*		*Knots.*	*Knots.*	*Knots.*
1	9	11	161	16.67	— 1.91	14.76
2	9	24	154	16.02	+ 2.15	18.17
3	9	36	141	14.85	— 2 35	12.50
4	9	53	134	14.21	+ 2.54	16.75
5	10	7	123	13.20	— 2.66	10.54
6	10	26	117	12.64	+ 2.74	15.30
7	10	44	104	11.41	— 2.76	8.65
8	11	2	97	10.73	+ 2.74	13.47
9	11	25	87	9.70	— 2.67	7.03
10	11	46	81	9.07	+ 2.55	11.62
11	12	14	71	8.00	— 2.34	5.66
12	12	43	61	6.89	+ 2.05	8.94

Note.—All curves of speed, &c., were drawn on a scale sufficiently large to enable speeds to be measured accurately to the nearest hundredth of a knot.

Observe that the trials are supposed to be begun about an hour after the turn of the tide. The worst time for measured mile runs is about slack water, for then the changes of tidal current are most violent and irregular in force and direction. It is not so much the speed of the current as the rapidity of change of speed that causes errors. From the apparent speeds in Table IV the two curves of apparent speed in Fig. 5 are drawn. Their mean curve, or curve of true speed, coincides so closely with the curve of true speed in Fig. 2, with which we started, that it is impossible to indicate the difference on the scale of the figure.

It is instructive to apply the same method of an imaginary trial to the "final mean" plan. Table V shows results of three series of four runs each, the errors in revolutions and intervals being distributed as in Table IV. Table VI shows the final mean speed as contrasted with the true mean speed corresponding to the

Table V.

No. of run.	Time at middle of run.	Revolutions.	True speed.	Speed of tidal current.	Apparent speed.
	h. *m.*		*Knots.*	*Knots.*	*Knots.*
1	9 11	161	16.67	— 1.91	14.76
2	9 24	163	16.84	+ 2.15	18.99
3	9 35	161	16.67	— 2.32	14.35
4	9 50	159	16 50	+ 2.51	19.01
5	10 06	114	12.36	— 2.65	9.71
6	10 25	117	12.64	+ 2.74	15.38
7	10 42	113	12.26	— 2.77	9.49
8	10 57	115	12.45	+ 2.76	15.21
9	11 23	69	7.78	— 2.68	5.10
10	11 45	72	8.11	+ 2.55	10.66
11	12 12	71	8.00	— 2.35	5.65
12	12 37	70	7.89	+ 2.10	9.99

mean revolutions. It is evident that the "final mean" method involves appreciable errors. The errors could be neglected if the four runs could all be made at the same number of revolutions. A glance at Table I shows that this is impracticable.

Table VI.

	Mean Revolutions.	Final Mean Speed.	True Mean Speed.
		Knots.	*Knots.*
First Series.....	161.00	16.724	16.67
Second Series.....	114.75	12.441	12.43
Third Series.....	70.50	8.003	7.94

It will be observed that in the preceding I have assumed the revolutions and the time on the course to be capable of exact determination. The instrument described by Assistant Engineer W. D. Weaver in the JOURNAL, Vol. III, No. 1, appears to be exact to all intents and purposes.

As it has been recommended for Service use by the Board on Construction, it will, I presume, be available for use in future progressive trials.

I consider that the method which I have outlined above is inherently more accurate, as regards speed, than the method of

runs in series, and am unable to see any serious practical difficulties in carrying it out.

One weak point lies in the fact that unless more than one card is taken during each run on the mile the spots for the I.H.P. curve are likely to be unfair.

The question is whether a mean curve through four unfair spots is liable to be more in error than a curve through a single spot which is the mean of four approximate observations. Personally, I should prefer to see the spots distributed.

But so far as my knowledge extends there is no practical difficulty in taking more than one indicator card while on the mile, even for fast runs. At 24 knots a ship is on the mile 2½ minutes, and at 10 knots 6 minutes. Two cards at the former speed and three at the latter would seem well within the possibilities. I should like to be favored with the opinions of some of the indicator experts of the Society upon this point.*

In a trial made in the manner which I have suggested it would be desirable to begin the runs at the top speed. It will be noted that if only one run be made at top speed it will be necessary to extend one of the curves of apparent speed beyond the last spot found on it. I would get over this difficulty by making three runs at top speed, thus obtaining a good closing spot for the curve of true speed.

Having made the top speed runs, it should be possible to make ten or twelve others, and complete the speed trials in about four hours in all, thus getting the whole of the mile trials over in the favorable part of one tide. The lowest speed I would suggest for mile trials would be eight knots. The speed results at lower speeds soon become unreliable.

The lower part of the speed and power curve is of use only for the determination of the "initial" or "constant" friction of the engine. This can, however, be obtained without trials on the mile, and I believe with more accuracy than by the method involving speed of ship.

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* Reference to the JOURNAL, Vol. II, p 419, shows that this estimate is perfectly reasonable, as more than eighteen double cards were taken in as many minutes under disadvantageous circumstances.—EDITOR.

Suppose we plot upon revolutions as base a curve of mean effective pressure reduced to the low pressure cylinder. Such a curve will approach the origin as indicated in Fig. 6. It will cut the axis of pressure, above the origin, horizontally. Its ordinate above the origin will represent the pressure equivalent to the initial friction of the engine, *i. e.*, the pressure required to keep the engine just moving against the friction due to tightness of glands, bearings, &c.

Now, let the I.H.P. of an engine be denoted by I, the mean pressure by p_m, and the revolutions by R, while the constant for horse-power is denoted by c.

Then, $$I = cp_mR.$$

Differentiating,

$$\frac{dI}{dR} = cp_m + CR\frac{dp_m}{dR}.$$

Let p_0 denote the mean pressure corresponding to initial friction.

Then, at the origin, $R = 0$, and

$$\frac{dI}{dR} = cp_0.$$

In other words the initial friction is measured by the inclination of the I.H.P. curve at the origin. The I.H.P. curve being plotted on revolutions, it is not necessary to observe speed to plot it.

So, after the speed trials on the mile were over, I would determine the power at various numbers of revolutions down to the lowest number which the engines could steadily maintain.

In plotting the curve to the origin we are met by the difficulty of plotting any curve with accuracy as to direction at the last known point.

This difficulty is overcome by laying off the I.H.P. curve negatively in the third quadrant, as shown in Fig. 6, thus obtaining a double curve, symmetrical in the first and third quadrants.

The origin is then a point of inflection of the double curve, and if a straight batten of uniform section is used we may be sure of obtaining a curve as nearly exact as the observations will admit. The inclination at the origin of the I.H.P. curve, or the

value of $\frac{dI}{dR}$, may be obtained by direct measurement. It is, however, more accurate to plot a curve of values of $\frac{I}{R}$ as indicated in Fig. 6.

By taking the spots on each side of the origin and sweeping in a fair curve we obtain the initial friction as accurately as by any other method. If preferred, instead of plotting a curve of $\frac{I}{R}$, a curve of mean pressure can be plotted at once by using the formula $I = cp_mR$.

The curves of Fig. 6 are for the *Yorktown*. Owing to the exceptionally small initial friction of the *Yorktown*, the inclination at the origin of the I.H.P. curve is much less than would usually be the case, and the curves of $\frac{I}{R}$ and mean pressure appear to curve more rapidly than if the initial friction were greater.

It is evident that the method of determining initial friction which I have outlined above may be applied to each engine of a twin or triple screw ship, and is not restricted to marine engines.

It appears to me that this method would be particularly useful for testing the initial friction of engines that are new, or that have been extensively repaired. A "progressive" dock trial can be made without much difficulty, and would be fully as valuable for the determination of the initial friction as a progressive trial under way. When made the ship should not be exposed to a current.

2. STANDARD TRIAL CONDITIONS.

Curves of speed and power will vary a good deal according to the conditions of the trial. Other conditions being the same, a ship with a foul bottom will develop less speed and more horsepower at a given number of revolutions than a ship that is thoroughly clean.

The displacement of the ship, the state of wind and sea, and the condition of the machinery, are all variable quantities and affect our curves by their variation. Trials of a ship under varied conditions would be most interesting and instructive, but for

trials that are made, or should be made, by all new ships it is evidently necessary to fix upon certain standard conditions.

The proper standard displacement for a speed trial is clearly what is called the Normal Load Displacement. This is a somewhat arbitrary displacement fixed for each ship by supposing that in addition to all fixed weights (hull, guns, machinery, &c.,) she carries definite quantities of coal, ammunition, provisions, &c.

As implied by the name, the normal load displacement should be about the average displacement of the ship during service.

What should be the standard conditions for trial as regards wind and sea, the bottom of the ship and the condition of the machinery?

Evidently they should be such that they can be easily reproduced for every ship independently of the individual judgment of the people in charge.

There can be very little question as to whether or not the water is smooth and the force of the wind slight, and it would be clearly impracticable to try all ships in water of a certain roughness with wind of a certain force. So we should evidently have as standard conditions smooth water and light wind.

Next, as to the condition of the bottom.

We have no such exact knowledge of the matter of fouling that we can say that the fouling of a ship's bottom has reached a given standard amount.

If we possessed such knowledge it would not always be practicable to wait for the ship to acquire the standard foulness. But we do know that, if a ship has just been cleaned and painted in dock, her bottom should be in a standard condition of smoothness, and a ship can be put in this condition at will.

So a clean bottom should evidently be adopted for a standard condition.

When we come to fix the condition of machinery on trial, the human factor necessarily comes in, since we cannot have "standard" men, in order to insure that every engine may receive exactly the same treatment. But it is evident that on trial the machinery should be in first-class condition.

I have pointed out in detail the reasons why progressive trials

are made under the conditions most favorable to speed inasmuch as we often hear such trials discredited because they show results which are not repeated during actual service.

A ship during ordinary cruising seldom shows the same speed for a given power as was shown over the measured mile, since cruising conditions are seldom the standard conditions which we have seen to be desirable for trial,

Progressive trials are not intended to show what a ship will do on service except in so far as they indicate the maximum results that may ever be anticipated.

They are intended—

First. To test the actual inherent maximum capacity of the ship as regards return of speed for various expenditures of power.

Second. To show the relative performance of a ship by comparison with results obtained from other ships.

Third. To afford data for use in connection with the designing of new ships.

Naval architecture can scarcely be called an exact science. This is especially true of that part of the naval architect's profession which deals with the question of the speed and power of ships. Advances in this branch are made step by step and must be based upon careful sifting and analysis of the performances of actual ships.

3. THE MEASURED MILE.

The course for progressive speed trials is usually made one nautical mile long. The nautical mile in this country is 6,080.27 feet.

For the determination of speed alone a shorter course might be used with the accurate Weaver register for recording time and revolutions on the course.

But determinations of speed and power are usually made simultaneously, and for a satisfactory determination of power a course nearly or quite a mile long is needed. Then, even at speeds of twenty knots or so, a ship is long enough on the course to enable three indicator cards to be taken from each cylinder—assuming plenty of observers available.

There are certain general considerations affecting the location of a measured mile, which it may be well to enumerate.

1. The mile should be so situated with respect to the coast or to islands that the range marks can always be clearly made out in good weather.

2. The situation should be such that the mile will be fairly well protected, insuring smooth water when the local weather conditions are favorable.

3. There should be plenty of room at each end of the mile. A ship at 20 knots, for instance, after turning and picking up the course range should have a straight run of about half a mile before coming on the measured course.

4. The mile should be so situated that the ships making runs over it need never cross or obstruct a channel or fairway that is much used.

5. The less current there is on the mile the better. It is much more important, however, that for the greater part of each tide the speed of the current shall change steadily—without sudden fluctuations—and that during that time the direction of the current shall be right on the steering range, so that the helm may be used as little as possible during any run.

6. A measured course should be close to a Navy Yard. Then a ship at the yard, just out of dock, could wait for a fine day, proceed to the course, run her progressive trials and get back to the yard the same day.

7. The depth of water on the mile should be sufficient. The question is—What is a sufficient depth?

We all know that the resistance of a ship increases in shoal water, this increased resistance being accompanied by marked changes in the appearance of the waves produced in the water by the motion of the ship. It is natural, then, to attack the question in hand by considering the effect of shoal water upon free waves. Rankine's "Trochoidal Theory" of waves, while not exact, does undoubtedly represent the phenomena and properties of actual water waves with sufficient approximation for every practical purpose.

Table VII, taken from a paper on "Water Waves," by the late

Table VII.

WAVES IN SHALLOW AND DEEP WATER.

Depth of water from mid-height of wave to bottom, expressed as a fraction of length of wave from crest to crest.	Ratio of Quantities for Shallow Water to Corrresponding Quantities for Deep Water.		
	Length and Velocity for Given Period.	Length for Given Velocity.	Velocity for Given Length.
.01	.063	15.90	.251
.02	.124	8.08	.352
.03	.186	5.376	.431
.04	.246	4.065	.496
.05	.304	3.289	.552
.075	.439	2.277	.663
.10	.557	1.796	.746
.15	.736	1.358	.858
.20	.847	1.180	.920
.25	.917	1.091	.958
.30	.955	1.047	.977
.35	.975	1.026	.987
.40	.987	1.013	.993
.45	.993	1.007	.996
.50	.996	1.004	.998
.55	.998	1.002	.999
.60	.999	1.001	.9999
.75	.9999	1.0001	.99999
1.00	.99999	1.00001	.999999

Naval Constructor Gatewood, shows clearly the relations between waves in shallow water and in water of unlimited depth.

It is evident that, if the depth of water is one-half the length of the wave, the wave is practically identical with that in water of unlimited depth. This, by the way, applies, not to a single wave or to a few waves, but to a system of an indefinite number of waves. There can be no doubt that the disturbance set up by such a system will extend as deeply into the water as the disturbance set up by a ship of any size yet constructed.

So we are led to the conclusion that a sufficient depth of water for a ship at a given speed is a depth *below the keel of the ship* equal to one-half the length of a wave traveling at the speed of the ship.

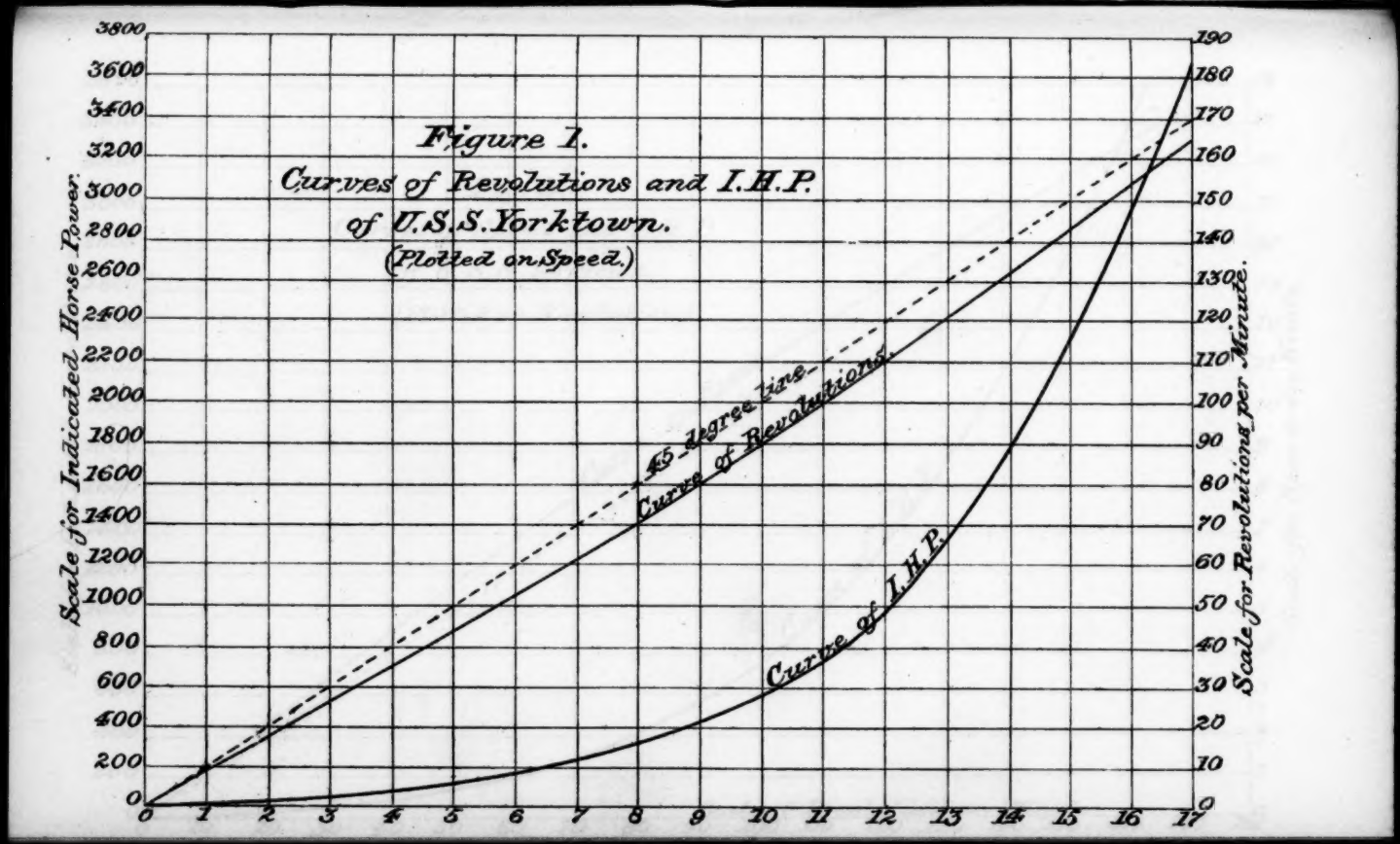


Figure 1.
Curves of Revolutions and I.H.P.
of U.S.S. Yorktown.
(Plotted on Speed.)
Scale for Indicated Horse Power.
Scale for Revolutions per Minute.
45 degree line
Curve of Revolutions
Curve of I.H.P.
0
200
400
600
800
1000
1200
1400
1600
1800
2000
2200
2400
2600
2800
3000
3200
3400
3600
3800
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
170
180
190
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17

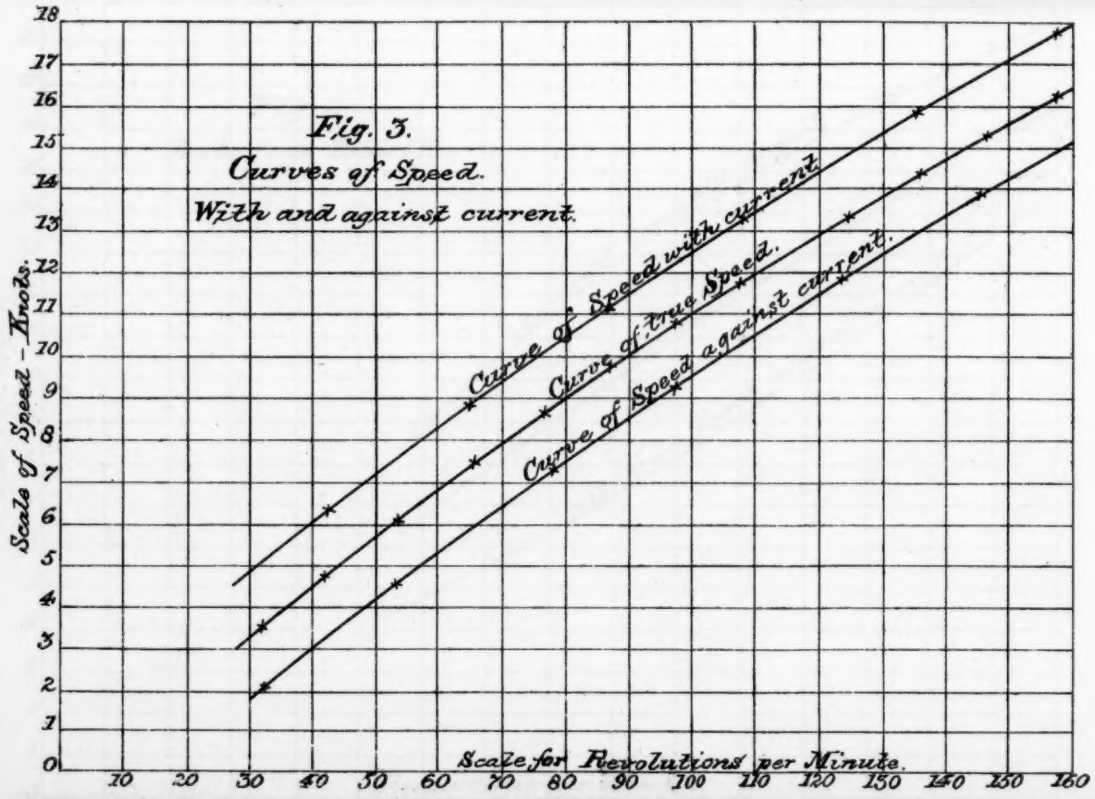


Fig. 3.
Curves of Speed.
With and against current.
Curve of Speed with current.
Curve of True Speed.
Curve of Speed against current.
Scale of Speed - Knots.
Scale for Revolutions per Minute.
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160

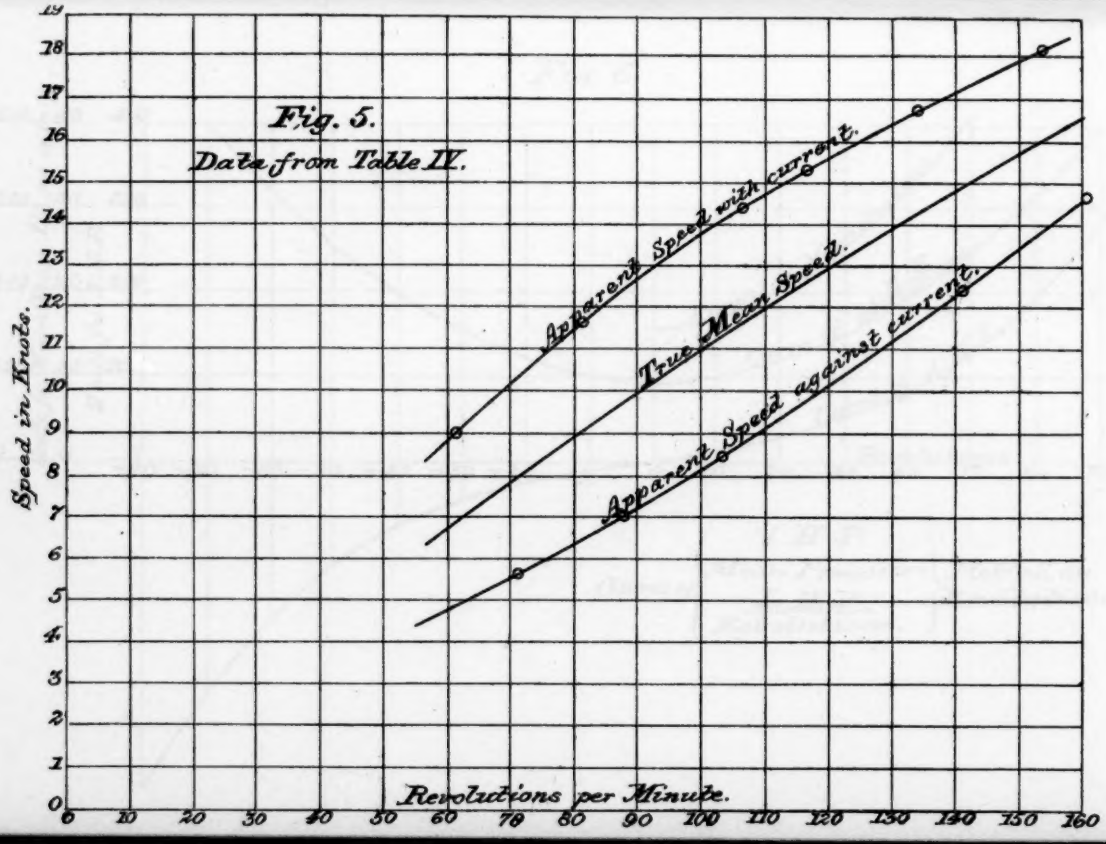


Fig. 5.
Data from Table IV.
Apparent Speed with current.
True Mean Speed.
Apparent Speed against current.
Speed in Knots.
Revolutions per Minute.
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
60
70
80
90
100
110
120
130
140
150
160

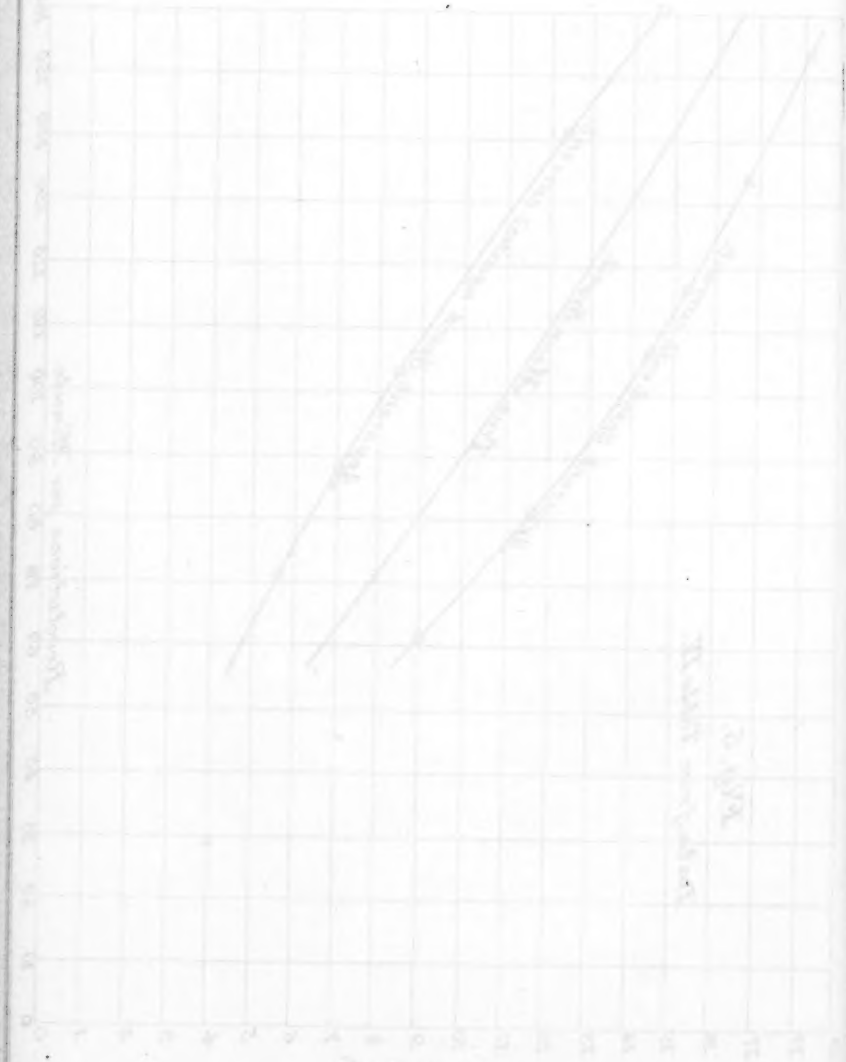
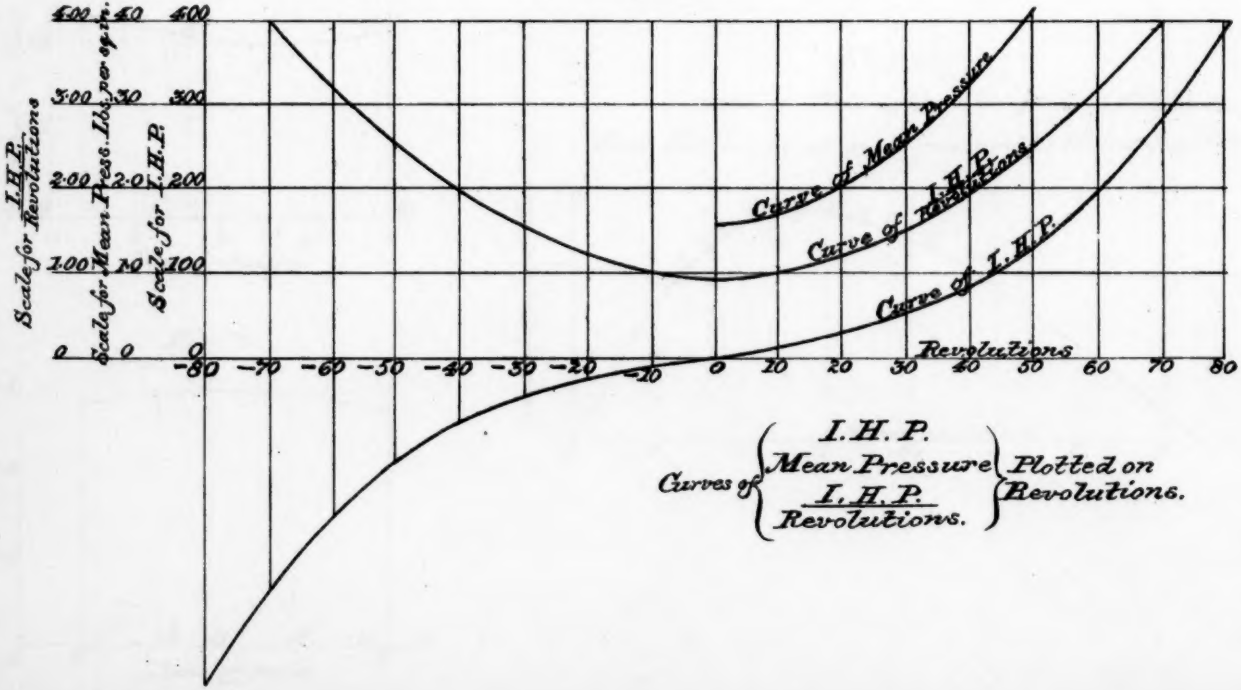


Fig. 6.



Curves of { I.H.P. / Mean Pressure ; I.H.P. / Revolutions. } Plotted on Revolutions.

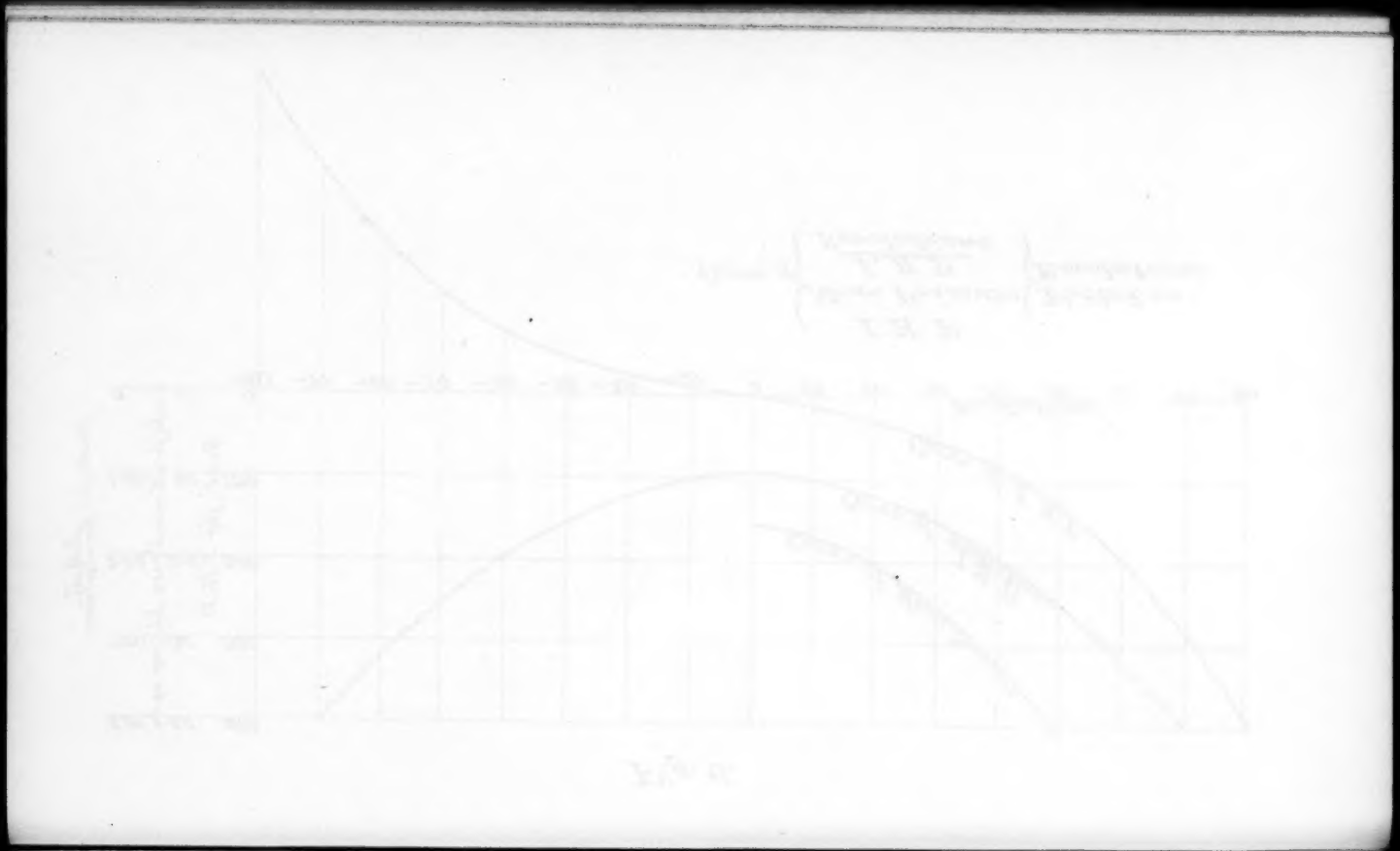


Fig. 2

Fig. 7.

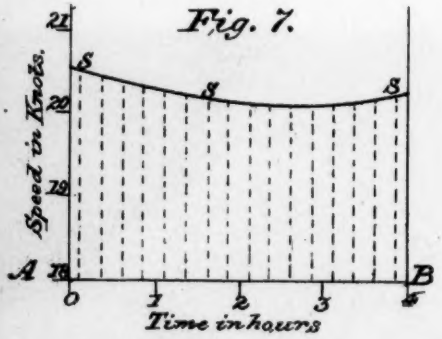


Fig. 8.

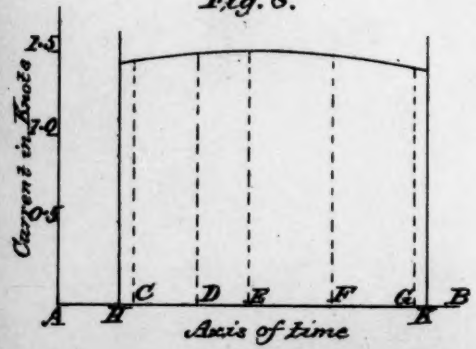
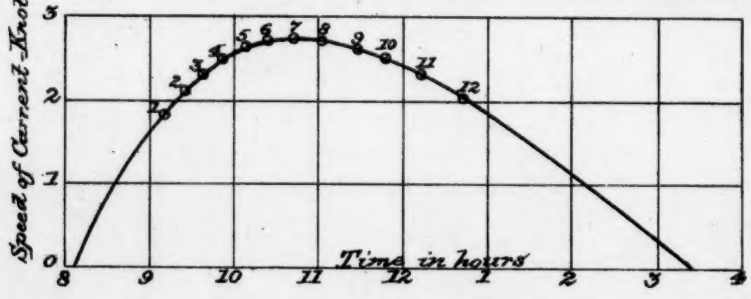


Fig. 4.

Curve of Tidal Current for One Ebb Tide.

Note: The numbered spots correspond to the runs of Table IV.



Now, if V denote the speed of a system of waves, in knots, and l the length from crest to crest, in feet, we have by the Trochoidal Theory, very approximately,

$$l = .56\ V^2.$$

Then the appropriate depth of water for a ship at speed V would be

$$\tfrac{1}{2}\ l \text{ or } .28\ V^2.$$

Speaking roundly, then, if V denote the highest speed of a ship on the mile, the depth of water should be such that *below the keel* we shall always have in feet $.3\ V^2$.

It is important that this depth be taken below the keel. Otherwise the formula will give too little depth for large slow vessels.

The above rule gives the following results for a few speeds:

Speed of ships.	Approximate depth below keel.
Knots.	*Feet.*
10	30
15	68
20	120
25	188
30	270

I am of the opinion that when this subject is more fully understood we shall find that smaller depths are permissible.

In deducing the above I have made ample allowances for the limitations of our knowledge upon the subject.

I have been unable to discover any record of reliable experiments upon the effect of shoal water upon a ship's resistance. Some reliable small scale experiments were made by the late Mr. Wm. Froude. The conclusions drawn from them were stated by his son, Mr. R. E. Froude, before the Institution of Naval Architects in discussing, at the spring meeting of 1892, a paper by Mr. W. H. White which touched upon the subject. Mr. Froude said: "In the Admiralty experimental tank at Torquay, a good many years ago—in fact, during my father's lifetime—some experiments were tried on models to determine the increase of resistance which was caused by shallow water. A false bot-

tom was fitted in the tank which could be set at various levels, so as to give various depths of water over it. The ships which were built in those days were not so fast as they are now. A 15-knot ship of 5,000 tons was a fast ship. Consequently the experiments which were made did not include such high speeds relatively to the depth of water submitted to experiment, as we now have to deal with, and the result of those experiments was this: We found that with water as shallow in proportion to the size of the model as the water at Stokes Bay in proportion to a ship of from 3,000 to 5,000 tons, there was a considerable increase of resistance—I think from 3 to 5 per cent.—and that was a nearly constant per centage at all speeds."

The water on the Stokes Bay mile is about 10 fathoms. By the rule given above a 15-knot ship, drawing 18 feet, should have 86 feet of water, or $14\frac{1}{3}$ fathoms.

It is a fact frequently lost sight of that the falling off in the speed of ships in shoal water is usually due not only to increased resistance but to diminished horse power. Suppose in deep water the engine is wide open showing its maximum mean effective pressure and maximum number of revolutions. Upon encountering shoal water and increased resistance, the revolutions almost invariably fall off, for reasons which I need not discuss. The engines being already wide open, the mean effective pressure cannot be increased by cutting off later. The horse-power would diminish directly as the diminished revolutions, were there not a slight increase of effective pressure because of diminished back pressure at the lower revolutions.

So far as I am aware, the measured mile in Narragansett Bay, near Newport, is the only one in this country. It is very fully and clearly described in the report of the officer under whose direction it was laid off. This description I give in an appendix.

4. CONTRACT SPEED TRIALS.

The question of four-hour contract speed trials was ably discussed in Vol. II, No. 1, of the JOURNAL of the Society, issued in February, 1890.

The *Philadelphia* and *San Francisco* have had speed trials since

that date, but there have been no such trials during the last two years.

As there are seven or eight ships now building whose contract speed trials should come off before 1894, a further discussion of the subject would seem to be timely.

In every case the contract speed requirement is, briefly, that a certain speed shall be developed at sea for four consecutive hours under conditions prescribed by the Secretary of the Navy.

Large premiums and penalties are involved, depending upon the speed developed.

The method of determining the contract speed which is, I believe, favored by a majority of the Society, is that proposed for the *Philadelphia* and *San Francisco* by the Engineer-in-Chief.

Briefly, this was to make preliminary trials on the measured mile in order to determine accurately the upper part of a speed curve such as shown in Fig. 2, and then make the four-hour run in the open sea, deducing the speed from the average revolutions.

Even supposing the curve of speed upon revolutions could be determined exactly by measured-mile trials, this method involves a slight error if the revolutions vary during the four-hour run. Owing to the fact that the curve of speed plotted on revolutions is not a straight line, the speed (as deduced from the speed curve) corresponding to the average revolutions is not the true average speed.

To determine the average speed with the utmost approximation by this method we should proceed somewhat as follows: Referring to Fig. 7, divide *A*, *B*, an axis of time representing the four hours of the trial, into a suitable number of, say, 15-minute intervals. From the revolutions recorded at each 15-minute interval and the curve of speed plotted from the results of the preliminary trials determine the mean speed during each 15-minute interval. Set it up on a suitable scale above the center of its interval, as indicated in the figure. The fair curve *S*, *S*, *S*, through the extremities of these speed ordinates will give the speed at each instant. Its area above the four-hour base line represents the distance traveled through the water by the ship in four hours.

The difference between the true average speed, deduced as

above, and the speed from the speed curve corresponding to the average revolutions will be negligible in a trial where the revolutions during the four hours do not vary much. But we must expect more or less variation of the revolutions during a four-hour run, and if the variation is as great in future trials as it has been in those already carried out, the speed corresponding to the average revolutions will be appreciably in error.

The actual four-hour trials of the *Philadelphia* and *San Francisco* were made back and forth over 40-mile courses, the tidal current on the course being supposed to be observed and allowed for.

There is but one correct principle to be followed in deducing the correction for tidal current, though it need not always be applied in precisely the manner which I shall describe.

Suppose that five ships anchored at intervals on the long course have noted the strength of tidal current at the passage of the trial ship.

Referring to Figure 8, on *A*, *B*, an axis of time, spot *C*, *D*, *E*, *F* and *G*, the times when the trial ship passed each of the five anchored ships as well as *H* and *K*, the times of entering and leaving the course.

Set off above *C*, *D*, &c., the corresponding speeds of current, observed as the trial ship was passing, and run a fair curve through the extremities of the ordinates. Then the area of this curve above *H*, *K*, represents the distance the current carried the ship during the run.

This distance being added to or subtracted from the known length of the course, we obtain the true distance made good by the ship through the water during the run. The time occupied by the run having been noted, the speed is readily deduced.

What are the principal objections to the revolution method of determining contract speed?

1. The trials required by this method are very elaborate. Hitches in carrying them out, or the interference of bad weather may make them very protracted.

2. No matter what care be taken the speed cannot be determined exactly on a measured course in a tideway, and the trial

"spots" for a speed curve plotted on revolutions will not fall on a perfectly fair curve.

The position of the correct true speed curve will then, to a certain extent, be a matter of opinion. Thus, if this method be used, we are liable to a conflict of "experts," and disagreement between the representatives of the contractors and those of the Department. It is not possible always to avoid such disagreements; but it is clearly best for all concerned to leave as little opportunity for them as possible.

3. There is a chance that, owing to bending of propeller blades or other accidents after the preliminary trials, they may have to be gone all over again before the final trial can be brought off.

What are the principal objections to the two-hour course, as it may be called, where one run is made in each direction during the four-hour trial?

1. It is difficult to get a satisfactory course.

2. For satisfactory current observations, the ships at anchor should be spaced not more than five miles apart. This would require eight or nine ships to cover a forty-mile course.

3. Instruments and methods used for measuring currents are more or less inaccurate.

All things considered, it appears that the only advantage of the first method over the second is its accuracy.

Now, the best method for contract trials is not necessarily that one which is most accurate. It is one which is simple, direct, easily carried out, and sufficiently accurate to insure substantial justice to both parties concerned.

The second method, with certain modifications, appears to me to fulfill the above requirements.

In conducting a four-hour contract trial I would propose—

1. That a one-hour course be used—*i. e.*, for a 20-knot ship a course 20 miles long. This would involve four runs and three turns. While turns are objectionable, if we grant that one is admissible (as on the two-hour course) the only additional objection to three appears to be the additional time they would require.

If suitable "conditions prescribed by the Navy Department"

are carried out by a competent Board, "bottling up" of steam and other jockeying off the course can be easily prevented.

2. For eliminating the effects of the current I would make observations of its strength, at a depth about five-eighths the mean draught of the trial ship, from five vessels anchored on the course at distances (from one end) of 2, 6, 10, 14 and 18 miles on a 20-mile course, similar spacing being adopted for courses of other lengths. With this arrangement the average current during a run would be, with ample approximation, the average of the five values obtained from the anchored ships as the trial ship passed.

Then all drawing of curves, &c., is avoided. A slight theoretical error is thus introduced for a single run, but it is so small as to be negligible, and the errors of runs with and against the current will be of opposite signs, and hence neutralize one another.

The weak point of the above method is due to the unavoidable inaccuracy of apparatus for determining speed of current. I understand, however, that the apparatus now used in the Coast Survey is very accurate. While individual current determinations might be slightly in error, it is reasonably certain that, owing to the number of the observations and the fact that runs would be made alternately with and against the current, the net resultant error would be negligible.

APPENDIX.

"U. S. Coast and Geodetic Survey Str. *Blake*,
"*Jamestown, R. I., July 23d, 1889.*

"DETAILS OF THE MEASUREMENT OF THE MILE FOR SPEED TRIALS OF THE NAVAL VESSELS IN NARRAGANSETT BAY.

"After making a compass reconnoissance of the locality, and selecting an approximate course of N. 0° 38′ E. (true) two range signals were established on Rose Island and well secured.

"Theodolite line was run from there to Conanicut Island,

striking the east shore a short distance north of the present ferry wharf.

"Measured 6,080.27 feet from this point, 90° from the line of range (nearly north), leveling every length. Two measurements disagreed three-quarters of an inch.

"The line was a third time measured roughly, to be sure of tally of lengths. Steel piano-forte wire was used under a uniform strain of twenty pounds. The upper-range signals were then established, and all four "cut in" by theodolite from Coast Survey triangulation points, the "cutting in" agreeing with the positions of the signals as established by measurement.

"The signals have four legs and pole, well bolted and braced, and painted white. The front one of each range is about 25 feet high, and has a diamond on the pole. The rear signals are about 35 feet high, and have a triangle on the pole.

"The legs of all are secured in about 500 pounds of rock and brick laid in Portland cement. Under the center of the pole of the lower signal is a mark, thus: A stone with a cross cut on its upper surface, buried about two feet under ground. On the top of this a bottle is placed, with the center of the neck in plumb, and buried under the surface about six inches. The pole of the rear upper range signal is over a large rock, near a stone wall on Mr. George W. Weedon's land. The mark under this pole is a cross cut on the surface and a hole drilled in the rock at the center of the cross. The front upper range-marks as the lower range, except that a hole is drilled in the rock instead of a cross.

"I send herewith the original letter from the persons having charge of or owning the property, giving authority to place the signals and keep them there.

"Course to be steered at right angles to the ranges is N. 0° 43′ E. (true). This passes in deep water 273 meters from the outlying rock off Fort Dumpling, and 215 meters from 25 feet curve; 460 meters from the shore line of Rose Island, when on the lower range, and 410 meters from 25 feet. Passes Taylor's Point (shortly before reaching the upper range) at a distance of 810 meters, and continuing above to Gould Island, it passes its western shore 160 meters distant; but this is less than 50 meters from 25 feet.

"The 'fore-and-aft' range is this: Above the beach below Fort Adams, and about ⅝ of a mile distant, is a bath-house with three doors on the water side, and steps leading down to a rocky beach. A short distance (about 100 feet) to the southward and westward of this bath-house is a tall signal pole. South (true) of this is a clump of trees, well marked with a group of houses to the westward. The signal pole is to be replaced by a large flag-pole by the owner of the estate on which it is placed.

"This pole, on with the right (westerly) slope of the clump of trees, is the course. The bath-house (which can be well seen) on with the left (easterly) slope of the clump is approximately correct (within a distance of two or three feet) on the course of the measured mile. Photographs of the ranges will be forwarded.

"On this range the depth of water to the southward of the measured mile is greater than 120 feet. Within the limits of the ranges there is one spot of 119 feet, near the upper one. To the northward the depth decreases to about 70 feet when abreast of the lower end of Gould Island. It is not advisable to trust the range above this point.

"Very respectfully,

"J. E. PILLSBURY,

"*Lieut. U. S. N., Ass't C. & G. Survey.*"

XXIII.

THE INFLUENCE OF SHOCK ON PROPELLER EFFICIENCY.

By Professor Wm. F. Durand, of Cornell University, Member.

In a paper published in the *Engineer* of January 11, 1867, and later in his miscellaneous papers, Prof. Rankine discusses the general subject of propeller efficiency. In the course of this paper there occurs the statement that the limit of the efficiency of a propeller is $\frac{v}{v + \frac{s}{2}}$, where v is the velocity of the ship, and s the change in the velocity of the water acted on. The conditions giving such efficiency are—

(1.) No transverse motion communicated to the water; (2) no friction between propeller and water; (3) a gradual change in the longitudinal velocity of the water acted on, from v to $(v + s)$, relative to the ship. In case (1) and (2) are fulfilled, but the water has its velocity changed suddenly from v to $(v + s)$, the efficiency is stated to become $\frac{v}{v + s}$, and the loss is laid to the influence of shock or to the eddies and whirls which accompany shock. The general case is provided for by supposing that there is always some loss of this nature, and that therefore the corresponding limit of the efficiency may be represented by $\frac{v}{v + (1 + c)\frac{s}{2}}$, where c may have any value from 0 to 1.

These ideas, in a more or less varying form, have been often repeated, and frequent references to the effect of shock on pro-

peller action are to be found. It has seemed to the writer that there are certain considerations whose bearing on this problem have not received due weight, and an examination of the question from a somewhat different standpoint may not be without interest.

We may first note that there are in general two methods of treating the action of a propeller: (1) By considering the *amount* of water acted on and the *change* effected in its motion by the action of the propeller, with the resultant reaction, work, etc. (2) By considering the resistance which the propeller meets with in passing through the water and endeavoring to find a law connecting this resistance with the total and useful work, etc. Each of these methods possesses its peculiar advantages, the first being the one usually employed, and the one used by Rankine. We shall here use the second method, which for questions of this character seems to possess certain advantages which will appear in the course of the discussion.

We will first give Rankine's proof of the main points in question:

(1) Suppose the velocity of action of the propeller upon the water to increase at a uniform rate from v to $(v + s.)$ The mean velocity with which the propelling apparatus has to work against the reaction R is then $\left(v + \frac{s}{2}\right)$ and the total work per second is $R\left(v + \frac{s}{2}\right)$, of which Rv is useful. Hence the efficiency in this case is $\frac{v}{v + \frac{s}{2}}$.

(2) Suppose the water to have its velocity changed suddenly from v to $(v + s)$ instead of gradually, as in (1). In this case the velocity of action of the propeller is $(v + s)$ throughout, and the total work per second is $R(v + s)$, of which Rv is useful as before. Hence the efficiency in this case is $\frac{v}{v + s}$.

In regard to this reasoning we observe that if the water could be acted on in the way stated, there could, of course, be no

objection to the results as given. This was presumably the assumption upon which the reasoning was based, without, however, implying that it could be actually attained in practice. The oar was mentioned in the paper as perhaps the most nearly fulfilling the ideal conditions required, and we shall therefore find it instructive to first examine from the other standpoint the action of the oar, or, what comes to the same thing slightly simplified, the action of a plane blade moving normal to itself and parallel to the direction of motion of the boat.

It should be noted that this difference in terms exists in the two methods of treatment. Velocities as used in the first method refer to the ship and to the velocity of the water acted on, both relative to the surrounding water taken as at rest. In the second method the velocities refer to the ship and to the velocity of the propeller estimated in a longitudinal direction, both likewise relative to the surrounding water considered at rest. In the case of the oar, and in fact throughout Rankine's paper, the velocity of the water and that of the propeller seem to be considered as the same. For the oar we will so consider it, then.

We will first note certain fundamental difficulties connected with the problem when treated by the first method as given in the paper.

The resistance experienced by a solid moving through a liquid is well known to depend on the velocity of motion, and for ordinary velocities to vary nearly as the square. The R in the equations above seems to be considered as constant for the time during which the action is considered. This would necessitate a constant velocity of action which is difficult to reconcile with the condition of a uniform increase. *Vice versa* a varying velocity of action necessitates a varying value of both R and v, and the reasoning by which the net and gross works are deduced seem hardly applicable to the case of the oar as usually operated.

Let us now proceed with the examination by the second method.

Let v be the velocity of the boat. Then the blade is supposed to increase its velocity relative to the boat from v_1 to $v_1 + s_1$ or relative to the water from 0 to s. Let a be the acceleration of

the blade, then the velocity relative to the water at any instant is at, counting t from the instant the action begins. Assuming for simplicity the resistance to vary as the square, we have as the resistance of the blade $R = Ba^2t^2$.

Let v denote the velocity of the boat at the same instant. Then $R = Cv^2$ and we have $v = at\sqrt{\frac{B}{C}}$ where B and C are constants depending on the form, size and surface of the blade and boat. The rate of work on the boat is $Rvdt = B^{\frac{1}{2}}C^{-\frac{1}{2}}a^3t^3dt$; $\therefore$ work done on boat in time $t = \frac{B^{\frac{1}{2}}C^{-\frac{1}{2}}a^3t^4}{4}$.

The rate of work on the water is $Ratdt = Ba^3t^3dt$; $\therefore$ work done on water is $\frac{Ba^3t^4}{4}$.

The first of these two works is useful, their sum is the total. We readily deduce for the efficiency the value

$$e = \frac{\sqrt{\frac{B}{C}}}{\sqrt{\frac{B}{C}} + 1} \quad \ldots\ldots\ldots\ldots \quad (1)$$

From the values above we have $\sqrt{\frac{B}{C}} = \frac{v}{at}$, $\therefore e = \frac{v}{v + at}$; or if we put s for at we have

$$e = \frac{v}{v + s} = \frac{v_1}{v_1 + s_1} \quad \ldots\ldots\ldots \quad (2)$$

Equation (1) shows the efficiency in the case to be independent of the velocities involved either of boat or blade, and, therefore, that it is constant no matter what such velocities may be. Equation (2) shows that the value of $\frac{v}{v + s}$ at any instant is also constant at the same value.

The method here used may be easily generalized as regards the special assumption made relative to acceleration and to variation as the square of the velocity. To this end let the blade so move that its velocity relative to the water is expressed as a function of the time by $s = at^p$, and let the resistance of bodies moving through water vary as the qth power of the velocity.

Then in the same way as above it is readily shown that e is independent of p and equals $\frac{\left(\frac{B}{C}\right)^{\frac{1}{q}}}{\left(\frac{B}{C}\right)^{\frac{1}{q}}+1}$, and that this expression is also equal to $\frac{v}{v+s}$, the quantities v and s being both measured at the same instant. The first value shows that e is independent of time and therefore constant throughout the stroke, and the second shows that the ratio $\frac{v}{v+s}$ has at all instants this same constant value. If the law governing the resistance of the blade is taken as different from that for the boat, an expression is easily found for the efficiency, but in its general form it is too complex to be of any great value. It would therefore seem to follow that no oar or propelling agent acting like one can have a limiting efficiency as high as $\frac{v}{v+\frac{s}{2}}$, where s signifies the velocity of the oar relative to surrounding still water, and if this velocity is taken as equal to that of the water acted on, then the limiting efficiency of the oar, as given in Rankine's paper, would seem to be greater than could be obtained for it under the conditions of gradual longitudinal change and no friction. The difficulty lies with the other condition, viz., that of "no transverse motion communicated to the water acted on." This condition the oar does not and cannot fulfill.

The method of treatment used above fulfills the condition of uniform change of velocity because the equations of motion are continuous in t, and the condition of no friction is fulfilled because the tangential forces acting on the blade could have no longitudinal component. The omission of the condition relative to no transverse motion, and the treatment from the other standpoint bring us, therefore, to a result differing widely from that given by Rankine. The water acted on by an oar or a blade such as has been supposed is, of course, far more than that comprised in a stream whose cross-sectional area is the area of the blade. The influence extends from the blade indefinitely, sup-

posing the water to extend so far; and the imparted velocities vary in direction and amount from the center of this indefinite stream outward. The total energy imparted to the water will therefore always include two components, a longitudinal and a transverse. The method we have used shows that at any instant the ratio between the total work spent on the water and that spent on the boat is exactly $\frac{s}{v}$, and this entirely independent of the relative size of the two components referred to above, or of the masses and water velocities which enter as factors into the differential portions whose sum makes up the energy imparted to the water. We therefore naturally reach for such a case a result differing from that given by the method used by Rankine.

We now come to the second of the points under discussion. It seems to be here assumed that the *water acted on* has its velocity suddenly changed from v to $(v + s)$ relative to the ship, and that, due to the accompanying "shock", there is a loss of energy with a consequent lowered efficiency. The important points here are involved in the ideas of (1) a sudden change of velocity, (2) consequent shock; and a careful examination of them will be necessary to our purpose. As to just what was meant in the original paper by a sudden change of velocity is not easy to determine. Probably not a change absolutely sudden, but one perhaps indefinitely approaching to such a limit.

We may first inquire what is meant mathematically by a sudden change of velocity. This is seen as follows: Let a particle so move that its velocity may be expressed generally by $v = f(t)$ where $f(t)$ is any continuous function of the time. The acceration will be given by $\frac{dv}{dt}$, and if, for any value of t, $\frac{dv}{dt}$ should become infinite, the particle would at that instant undergo a sudden change of velocity. The force necessary at each instant is $m\frac{dv}{dt}$, and, if m is finite, the force at the instant in question must be ∞. The sudden acceleration or change in the velocity of a body will therefore require, at least for an infinitely short time, the existence of an infinitely large force. This does not seem admissible,

and we therefore conclude that no finite portion of matter, large or small, can change its velocity with absolute suddenness. This is, indeed, almost or quite self evident, for if we consider the actions between different bodies as they occur about us we readily see that in no case are they associated with absolutely sudden changes of velocity. All motions are and must be gradual. It is not to be denied, of course, that changes of motion may be conceived of which indefinitely approach the condition of absolute suddenness, and the question of its existence at the limit is largely a matter of the definition of terms. With that we are not concerned. Our purpose is simply to point out that there is no discontinuity in the law according to which matter undergoes changes of velocity, and therefore no discontinuity in the law connecting energy developed and work done. It may also be noted that changes which to us appear sudden are so in relative degree only.

We shall find it instructive to examine in this light the action of a bat on a base ball. The impact between the two destroys the motion of the ball in one direction and gives it motion in the opposite. However rapid these motions apparently are, it is sure that no portion of the ball has changed velocity with absolute suddenness. The impact was accompanied by mutual distortion of the bat and ball. These were necessarily gradual and finally brought the particles of the ball to rest (not necessarily all at the same instant), and then as gradually gave it motion in the opposite direction.

If, now, we could measure the total energy spent on the ball and compare it with the kinetic energy of the ball as a whole just as it leaves the bat, we should find the former greater than the latter. There is, therefore, an apparent loss of energy. We know, however, that this loss exists as heat and general vibration of the ball, the latter gradually becoming transformed into heat. With reference to the ball, we may call this heat energy internal, and the energy of translation external, each representing a part of the total work spent upon it. The loss represented by the internal energy is due, therefore, to the fact that molecular and material vibrations naturally accompany the action between the

bodies concerned. In general it may be stated that when one solid acts on another, such action being accompanied by a change in their relative velocities, there will usually be a loss of external energy, such loss being balanced by an equal gain in internal energy. The amount of such loss will depend on the form and nature of the bodies and their velocities. It is readily seen, also, that the more nearly sudden the changes in velocity, the greater, relatively, will be the energy passing into this internal form. In theoretical mechanics on the subject of impact, reference is usually made to bodies with a coefficient of restitution equal to 1, for which such loss is 0. Such bodies are, however, imaginary, and it follows in actual practice that there always is such loss. While its existence is thus well established, its quantitative measure, either by calculation or experiment, is, in the general case, impracticable. General considerations would seem to indicate, however, that in most cases it would be small relative to the external kinetic energy involved, and that only in extreme cases would it reach an amount equal to it. With a proper understanding of the term it is not improper to state this loss as due to *shock*, referring to the impact or shock between the bodies concerned, but without assuming anything as to a sudden variation in velocity of either body, or of any part thereof.

Turning, now, from the subject of solid or rigid bodies, we may ask whether similar phenomena are to be met with in the case of liquids acted on by solids, or *vice versa*. We may first assert, from general principles, that no liquid particle or mass can have its velocity changed with absolute suddenness. We then note that the water acted on by a solid around which it flows may be divided into two portions: First, That of which the particles follow open stream lines. Second, That which is thrown into eddies and whirls, the particles of which move in interlacing spirals and loops.

With regard to the first portion we note that the stream lines are smooth curves, that is, curves without a sudden change of direction, or with no discontinuity of the first derivation. This follows from their general equations for perfect liquids, and the addition of viscosity can introduce no change in this particular.

If, then, such is the character of stream lines in the case of a fixed solid with the liquid flowing past, it follows for the case of a solid moving through the liquid, the relative conditions being the same, that the motion of a particle as it moves out of the way and then back again, must be gradual and smooth. A sudden change in the velocity of such a particle would imply a sudden change in the direction of the stream line, and this, as we have seen, is inadmissible.

Turning now to the second portion of the water, that thrown into eddies, it is equally sure from general principles that no particle changes velocity with absolute suddenness.

The term *shock*, as generally used by writers on hydrodynamics, is associated with the formation of such eddies and whirls, and the loss of energy consequent upon their existence. Where mention is made of a sudden change of velocity reference must be intended, as we have seen, to a change of, perhaps, great rapidity, but by no means of absolute suddenness. The subject is usually illustrated by the phenomena occurring in a pipe carrying a liquid, at a point where the cross section suddenly enlarges. We can readily imagine a sudden enlargement in a pipe. It is a geometrical conception, but dynamic actions do not and cannot take place with absolute suddenness. They, as a part of the physical world, are conditioned in time as well as in space. But to return to our pipe, it is shown in such cases by reasoning and supported by experiment, that there is a loss of energy at such points, accompanied by eddy formation. It therefore results that, at a point well beyond the place of enlargement, the total energy per pound of water is less than its value before reaching the enlargement. It is to be remembered that in any mass of water in steady motion the total energy may be divided into three parts: (1) The kinetic energy, due to the velocity. (2) The potential energy, due to the height or head above some datum plane. (3) The potential energy, due to pressure. In the case of the pipe with sudden enlargement, supposing it horizontal, there is a loss of kinetic energy and a gain of pressure energy, the energy of head remaining unchanged. The loss is, however, greater than the gain, and it is readily shown by rea-

soning and supported by experiment that the amount of the loss per pound of water is measured very closely by the kinetic energy due to a velocity equal to the change in velocity in passing from one part to the other. That is, if v_1 and v_2 are the velocities before and after passing the enlargement, then the loss is $\frac{m}{2}(v_1-v_2)^2$. (See Encyc. Brit. article "Hydromechanics," or Résal, Mécanique Général, Vol. II, p. 298.

The case of a solid moving in an indefinite body of water is, however, far more complex. It is equivalent to the action of an indefinite stream flowing about an obstacle. It is certain that, except in special cases, eddies will be formed. We may consider furthermore, that the kinetic energy involved in such eddy motion has much the same relation to the kinetic energy of the stream as a whole, that the energy of internal agitation in the case of the ball has to the energy due to its velocity of translation. That is, if for a stream, the three-fold energy referred to above be considered as external energy, we may properly consider the energy in the eddy as internal, more especially as it rapidly becomes transformed into heat. It follows, then, so far as external energy is concerned, that the gain in energy of a stream may be less than the work spent upon it, the result being that there is a real loss of available energy. In comparing this case with that of the pipe, we note that, in the former, the stream being indefinite in extent, the water acted on has its velocity changed by varying amounts, according to its location relative to the solid, so that the expression, "change of velocity," when applied to the action as a whole, is of extremely vague meaning.

The "energy due to change of velocity" is, of course, equally indefinite when applied to the stream as a whole. It is, of course, not to be denied that, by making certain suppositions somewhat similar to those used in the treatment of the case of the pipe, a value for the loss might be found by an integration extending over the indefinite cross-sectional area of the stream. The suppositions made for the case of the pipe are, however, uncertain *a priori*, and therefore depend upon experiment for confirmation.

In the case of the indefinite stream there seems no authority *a priori* for extending the suppositions used in the case of the pipe, or for making any others which will give a direct solution. Any suppositions which might be made, therefore, would be subject to experimental confirmation, which, from its very nature, would be an extremely difficult operation. While, therefore, the exact determination of this loss is a matter of great difficulty, we may say in general that it will depend on the surface form of the body as influencing the stream line forms, as well as on its velocity and location relative to the surface, and on the characteristics of the liquid.

It is proper to add that if the body has such a simplified form that the equations of its stream lines may be derived mathematically, then the energy involved may be approximated to. (See Craig, " Motion of a solid in a liquid.") Such cases, while mathematically interesting, are, however, of no practical importance in the present inquiry, as the circumstances are such that no equations, even measurably accurate, could be derived for the stream lines.

To return now to the main points, we remember that the maximum loss due to shock is given by Rankine as $\frac{mv^2}{2g}$, where m is the water acted on and v is a motion of translation, the longitudinal change in the velocity of the water. But we have endeavored to show that the loss due to shock, if the term is to be used at all, can be understood to refer only to the energy involved in the production of eddies. The water involved in these eddies is by no means all that is acted on by the propeller, and, whether the portion be greater or less, the eddy motion has little to do with the useful change of momentum from which the thrust is derived. The longitudinal component of the motion of translation is that from which the thrust is mostly obtained, and neither this motion nor any other can be produced with absolute suddenness, nor can there be absorbed by it any more energy than the equivalent of that due to its own velocity. If these points are well taken, it would seem to follow that this " eddy" loss, whatever it may be called, would in no case have so simple a

value as $\frac{mv^2}{2g}$, nor would its complete avoidance have the result of raising the limiting efficiency from $\frac{v}{v+s}$ to $\frac{v}{v+\frac{s}{2}}$. In fact, it is not difficult to see that if the oar considered above were given such a form as to completely avoid eddies, yet the efficiency would still remain $\frac{v}{v+s}$, exactly the same as with the flat blade assumed, with which eddies would be formed. In this case, therefore, there is absolutely no gain in efficiency due to the avoidance of eddies, always supposing the sizes to be such that the values of v and s are the same in each case. This, at first sight, may seem surprising, but a little study into the formation of the eddies in such a case furnishes the explanation. The eddies are formed first at the edge of the blade, and the motion of rotation given them is there longitudinal in direction. It follows that their formation presents a direct resistance to the blade, the same as would any other longitudinal change in momentum, and that they therefore contribute directly to the resistance whose reaction furnishes the thrust. Furthermore, the existence of eddies is accompanied by an imperfect closing in of the stream lines behind the oar, and this results in a net normal pressure directed from aft forward. This net normal pressure is, in fact, the thrust, and from the very nature of the case it must be normal to the blade, and therefore parallel to the direction of action of the working agent. The whole question turns on this last fact, for under such conditions no work can be expended without receiving in return a proportional thrust. This result is readily generalized as follows:

Let the propelling agent be a body of any form whatsoever—symmetrical or not—about a longitudinal plane or axis. Let its direction of motion be *longitudinal*. The body may be considered at every point of the surface as subjected to two forces, one normal, directed inward, the other a tangential force due to frictional resistance. The total result of the forces may be represented by three components; one vertical, one transverse, and

one longitudinal. Since the motion is longitudinal, there is no work done with reference to the first two. The third is equal to, or in fact is, the thrust, and is directly opposed to the motion of the body, and therefore to the action of the machine from which comes the supply of work. This resistance is overcome at a velocity of $(v + s)$ and the work spent on the body is therefore $R(v + s)$. Of this Rv is useful and the efficiency is therefore $\frac{v}{v+s}$. This result, it is seen, is entirely independent of the existence or non-existence of friction or of eddies, and, therefore, in such a case, nothing which can be done to the form or surface of the body will in any way affect the efficiency—always supposing that the v and s remain the same.

In the case of the ordinary screw propeller the motion is not longitudinal, and the existence of eddies entails the performance of a certain amount of work for which no equivalent thrust is obtained. This is, of course, prejudicial to efficiency, and it is therefore desirable in such cases to avoid their formation as far as possible. As to what their quantitative effect is, however, we have little data whereon to form an estimate. Neither are we much better off as regards the form of blade which will best avoid their formation. One thing of prime necessity to the intelligent study of such a problem is a complete knowledge of the direction of the stream lines as they come to the propeller. That they are not in general longitudinal is well known, but until their direction can be well mapped out for all parts of the screw disk area, little can be done in the way of an intelligent modification of the blade looking toward eddy decrease. The problem is further complicated by the fact that these directions are dependent not only on the ship and her velocity, but also on the form and revolutions of the propeller, so that at best it is a question whose solution is implicated within itself, and to which only approximations can be made. In any case it does not seem likely that eddy formation can be wholly prevented, and it is perhaps doubtful whether, in well-designed propellers, it is of sufficient amount to warrant any very great changes looking to its avoidance—changes which might lead in some other way to losses greater than those which they were intended to correct.

XXIV.

TESTS OF RIVETED JOINTS MADE AT THE WATERTOWN ARSENAL FOR THE BUREAU OF STEAM ENGINEERING, NAVY DEPARTMENT.

Most of the experiments on the strength of riveted joints have been made on a comparatively small scale, the width of the specimen generally comprising not more than two pitches, and this has led to the idea that possibly the results were not truly indicative of what might occur in a long seam of the same style.

Again, most experiments have been made on thin plates and on single or double-riveted seams, while the high pressures and large diameters now in use have brought in thick plates and treble riveting.

In view of these facts, the Bureau of Steam Engineering decided to take advantage of the exceptional facilities offered by the Emery testing machine at the Watertown Arsenal, and to carry out a series of tests on specimens of riveting which should be sufficiently large to be a true indication of the whole seam, while the plates should be of considerable thickness and the styles of riveting should include the strongest forms.

The specimens were prepared at the Brooklyn Navy Yard early in 1887, and the tests were made at Watertown in the same year. The results have been published by the Ordnance Department of the Army in "Tests of Metals for 1887"; but as this document does not reach a large part of those who read the JOURNAL, it has been considered advisable to place the record of these valuable tests where it would be more accessible.

The joints were made of open-hearth steel plates, which had drilled rivet holes and sheared edges. Both steel and wrought-iron rivets were employed, as noted in the detailed account of the tests. They were driven by a steam riveting machine.

The strips tested to show the quality of the metal employed in the joints were furnished by the steel makers as being the same grade of metal, although not from the same sheets as the joints.

The tests were conducted in accordance with the general practice at Watertown.

It will be seen that the thickness of the plates varied considerably at different edges. These measurements are given on the illustrations of the joints in their respective places.

After testing from a number of the joints the rivet heads were planed off, the rivets driven out, and butt straps removed, thus giving access to the rivet holes in the plates, which were then measured and their elongation recorded on the sketch. From these figures an idea may be formed with regard to the transmission of the stresses to the different rows.

The metal drew down in thickness in zig-zag lines from rivet-hole to rivet-hole of adjacent rows, in spite of the fact that the extent of metal thus traversed was greater than directly across the plate through one line of rivet holes.

The average tensile strength of the three specimens of each thickness of plate was used in computing the efficiencies of the joints.

It will be noticed that the efficiencies of joints I_1, I_2, I_3 are not given, nor the tensile strength of the plate. This is because no specimens of this thickness of plate (nine-sixteenths inch) were tested for tensile strength.

TENSILE TESTS OF THE MATERIAL.

TEST NO. 904.—SPECIMEN C_1.—THICKNESS, **FIVE-EIGHTHS** INCH.

Gauged length, 15 inches; cross section, $12'' \times 0.634''$; area, 7.608 square inches; plate warped.

Applied loads.		In gauged length.		Remarks.
Total.	Per square inch.	Elongation.	Set.	
Pounds.	*Pounds.*	*Inches.*	*Inches.*	
7,608	1,000	0.0000	0.0000	Initial load.
38,040	5,000	.0024	.0000	
76,080	10,000	.0054	.0001	
114,120	15,000	.0083	.0002	
152,160	20,000	.0114	.0007	
159.768	21,000	.0122	.....	
167,376	22,000	.0130	.....	
174.984	23,000	.0140	.....	
182,592	24.000	.0148	.....	
190,200	25,000	.0160	.0027	
192,000	.....	.0165	.....	
194,000	.....	.0170	.....	
196,000	.....	.0173	.....	
198,000	.....	.0177	.....	
200,000	.....	.0183	.....	
202,000	.....	.0185	.....	
204,000	.....	.0188	.....	
206,000	.....	.0191	.....	
208,000	.....	.0195	.....	
210,000	.....	.0199	.....	
212,000	.....	.0205	.....	
214.000	.....	.0210	.....	
216,000	.....	.0218	.....	
218,000	28,650	.0225	.....	Elastic limit.
220,000	.....	.0238	.....	
222,000	.....	.0285	.....	
224,000	.....	.0540	.....	
226,000	.....	.1450	.....	
228,240	30,000	.20	.....	
235.848	31,000	.23	.....	
243.456	32.000	.26	.....	
251,064	33,000	.28	.....	
258,672	34,000	.29	.....	
266,280	35,000	.31	.....	
273.880	36.000	.35	.....	
281.496	37,000	.39	.....	
289,104	38,000	.42	.....	
296.712	39.000	.46	.....	
304.302	40 000	.50	.....	
311,928	41,000	.55	.....	
319,536	42,000	.60	.....	
327,144	43,000	.65	.....	
334,752	44,000	.72	.....	
342,360	45,000	.81	.....	
349,968	46,000	.86	.....	
357.576	47.000	.95	.....	
365,184	48,000	1.04	.....	
372,792	49,000	1.16	.....	
380.400	50,000	1.27	.....	
388,008	51,000	1.47	.....	
395,616	52,000	1.67	.....	
403.224	53,000	2.17	.....	
410,832	54,000	2.85	.....	
411,700	54,120	3.20	.....	Tensile strength.
			5.23	= 34.9 per cent.

Elongation of inch sections: .15", .17", .20", .21", .25", .26" .34", .42", .58", *1.20", .39", .33", .30", .22", .21".

Area at fracture: $9.76'' \times .45'' = 4.392$ square inches. Contraction, 42.2 per cent.

Appearance of fracture, silky, slightly lamellar. Fracture open at middle, .50"; at edges, .10" and 0.

TEST NO. 905.—SPECIMEN C_2.—THICKNESS, **FIVE-EIGHTHS** INCH.
Gauged length, 15 inches; cross section, $12'' \times .639''$; area, 7.668 square inches.

Applied loads.		In gauged length.		Remarks.
Total.	Per square inch.	Elongation.	Set.	
Pounds.	*Pounds.*	*Inches.*	*Inches.*	
7,668	1,000	0.0000	0.0000	Initial load.
38,340	5,000	.0017	0.0000	
76,680	10,000	.0042	0.0000	
115,020	15,000	.0065	0.0000	
153,360	20,000	.0089	0.0000	
161,028	21,000	.0094	.....	
168,696	22,000	.0098	.....	
176,364	23,000	.0103	.....	
184,032	24,000	.0108	.....	
191,700	25,000	.0112	—.0001	
194,000	.....	.0113	.....	
196,000	.....	.0115	.....	
198,000	.....	.0116	.....	
200,000	.....	.0118	.....	
202,000	.....	.0120	.....	
204,000	.....	.0123	.....	
206,000	.....	.0125	.....	
208,000	.....	.0127	.....	
210,000	.....	.0130	.....	
212,000	.....	.0133	.....	
214,000	.....	.0136	.....	
216,000	.....	.0140	.....	
218,000	.....	.0144	.....	
220,000	.....	.0150	.....	
222,000	28,950	.0154	.....	Elastic limit.
224,000	.....	.0162	.....	
226,000	.....	.0170	.....	
228,000	.....	.0183	.....	
230,000	.....	.0213	.....	
232,000	.....	.0277	.....	
234,000	.....	.0480	.....	
236,000	.....	.0995	.....	
237,708	31,000	.2175	.....	
245,376	32,000	.23	.....	
253,044	33,000	.25	.....	
260,712	34,000	.28	.....	
268,380	35,000	.31	.....	
276,048	36,000	.33	.....	
283,716	37,000	.37	.....	
291,384	38,000	.41	.....	
299,052	39,000	.45	.....	
306,720	40,000	.50	.....	
314,388	41,000	.54	.....	
322,056	42,000	.59	.....	
329,724	43,000	.66	.....	
337,392	44,000	.71	.....	
345,060	45,000	.78	.....	
352,728	46,000	.86	.....	
360,396	47,000	.95	.....	
368,064	48,000	1.07	.....	
375,732	49,000	1.21	.....	
383,400	50,000	1.47	.....	
391,068	51,000	1.58	.....	
398,736	52,000	1.80	.....	
406,404	53,000	2.08	.....	
414,072	54,000	2.98	.....	
414,800	54,100	3.42	.....	Tensile strength.
0	0	0	5.52	= 36.8 per cent.

Elongation of inch sections: .23", .20", .24", .26", .34", .35", .40", *1.03", .69", .48", .35", .31", .26", .21", .17".

Area at fracture, $9.62'' \times .48'' = 4.618$ square inches. Contraction, 39.8 per cent.

Appearance of fracture silky, slightly lamellar. Fracture open at middle, .38", at edges, .10" and 0.

TEST NO. 906.—SPECIMEN C_8.—THICKNESS, **FIVE-EIGHTHS** INCH.

Gauged length, 15 inches; cross section, 12″ × .672″; area, 8.064 square inches.

Applied loads.		In gauged length.		Remarks.
Total.	Per square inch.	Elongation.	Set.	
Pounds.	*Pounds.*	*Inches.*	*Inches.*	
8,064	1,000	0.0000	0 0000	Initial load.
40,320	5,000	.0018	0.0000	
80,640	10,000	.0042	0.0000	
120,960	15,000	.0068	.0002	
161,280	20,000	.0100	.0010	
169,344	21,000	.0108	.....	
177,408	22,000	.0115	.....	
185,472	23,000	.0123	.....	
193,536	24,000	.0133	.....	
196.000	.....	.0137	.....	
198,000	.....	.0141	.....	
200,000	.....	.0144	.....	
201,600	25,000	.0148	.0029	
204,000	.....	.0152	.....	
206,000	.....	.0155	.....	
208,000	25,790	.0158	.....	Elastic limit.
210,000	.....	.0168	.....	
212,000	.....	.0176	.....	
214,000	.....	.0188	.....	
216,000	.....	.0218	.....	
218,000	.....	.0570	.....	
220,000	.....	.0740	.....	
222,000	.....	.0987	.....	
225.792	28,000	.1800	.....	
233,856	29.000	.21	.... .	
241.920	30.000	.23	.....	
249,984	31,000	.25	.....	
258,048	32,000	.27	.....	
266,112	33,000	.29	.....	
274,176	34,000	.33	.....	
282,240	35,000	.36	.....	
290,304	36,000	.38	.....	
298,368	37,000	.42	.....	
206,432	38,000	.46	.....	
314,496	39.000	.51	.....	
322,560	40,000	.55	.....	
330,624	41,000	.60	.....	
338,688	42.000	.65	.....	
346,752	43,000	.71	.....	
354.816	44.000	.78	.....	
362.880	45.000	.85	.....	
370.944	46,000	.94	.....	
379,008	47.000	1.07	.....	
387,072	48,000	1.18	.....	
395,136	49,000	1.30	.....	
403.200	50,000	1.48	.....	
411,264	51,000	1.71	.....	
419.328	52,000	2.13	.....	
426,600	52,900	2.98	.....	
426,500	.....	3.18	.....	
426,200	.....	3.32	.....	
426,000	.....	3.42	.....	
425,000	.....	3.62	.....	
424.000	.....	3.75	.....	
423,000	.....	4.00	.....	
422,000	.....	4.05	.....	
420,000	.....	4.40	.....	
415,000	.....	4.63	.....	
410,000	.....	4.82	.....	
395,000	.....	.....	.....	
280,000	.....	.....	.....	
0	0	0	5.28	

Elongation of inch sections: .17″, .21″, .28″, .37″, .50″, *1.23″, .51″, .41″, .34″, .27″, .26″, .22″, .16″, .17″, .18″. Area at fracture, 9.65″ × .49″ = 4.728 square inches. Contraction, 41.3 per cent. Appearance of fracture silky, lamellar. Fracture open at middle, .45″, at edges, .10″ and 0. The maximum contraction in thickness, after passing the highest load and before rupture occurred, took place along an oblique line, making an angle of 53° with the axis of the specimen. Rupture occurred in a line nearly at right angles to the axis of the specimen, intersecting the line of least thickness of metal at about the middle of the plate.

10.50″ × .585″
10.45″ × .580″
10.23″ × .567″
10.08″ × .555″
9.97″ × .534″
} Minimum width and thickness.

Opened fracture at middle of plate.

= 35.2 per cent.

*Crack opened about 4″ long by 10″ wide at middle. Specimen now struck several blows in vicinity of fracture with 15-pound sledge without apparent injury. Rupture now completed by continuing the stretch of the plate.

TEST NO. 907.—SPECIMEN F_1.—THICKNESS, **SEVEN-EIGHTHS** INCH.
Gauged length, 15 inches; cross section, 8.51″ × .882″; area 7.5 square inches.

Applied loads.		In gauged length.		Remarks.
Total.	Per square inch.	Elongation.	Set.	
Pounds.	*Pounds.*	*Inches.*	*Inches.*	
7,500	1,000	0.0000	0.0000	Initial load.
37,500	5,000	.0019	0.0000	
75,000	10,000	.0043	0.0000	
112,500	15,000	.0065	0.0000	
150,000	20,000	.0090	.0001	
157,500	21,000	.0095	.....	
165,000	22,000	.0101	.....	
172,500	23,000	.0106	.....	
180,000	24,000	.0115	.....	Elastic limit.
187,500	25,000	.0129	.0017	
190,000	.....	.0139	.....	
192,000	.....	.0149	.....	
195,000	26,000	.0164	.....	
197,000	.....	.0186	.....	
199,000	.....	.0241	.....	
202,500	27,000	.1540	.....	
210,000	28,000	.17	.....	
217,500	29,000	.19	.....	
225,000	30,000	.22	.....	
232,500	31,000	.24	.....	
240,000	32,000	.26	.....	
247,500	33,000	.28	.....	
255,000	34,000	.32	.....	
262,500	35,000	.36	.....	
270,000	36,000	.40	.....	
277,500	37,000	.45	.....	
285,000	38,000	.49	.....	
292,500	39,000	.54	.....	
300,000	40,000	.59	.....	
307,500	41,000	.66	.....	
315,000	42,000	.71	.....	
322,500	43,000	.79	.....	
330,000	44,000	.86	.....	
337,500	45,000	0.95	.....	
345,000	46,000	1.06	.....	
352,500	47,000	1.19	.....	
360,000	48,000	1.33	.....	
367,500	49,000	1.59	.....	
375,000	50,000	1.94	.....	
382,500	51,000	2.34	.....	
386,400	51,520	3.25	.....	Tensile strength.
0	0	.....	5.08	= 33.9.

Elongation of inch sections: .17″, .17″, .25″, .29″, .34″, .46″, *1.31″, .49″, .34″, .30″, .24″, .24″, .20″, .17″, .11″.

Area of fracture: 6.53″ × .63″ = 4.11 square inches. Contraction 45.2 per cent.

Appearance at fracture, silky, slight lamination.

Fracture opened at middle of width of plate, and extended to the edges. After fracture, placing the ends together in line, there was an opening at the middle of .44″, at one edge .10″, the other end being closed.

TEST NO. 908.—SPECIMEN F_2.—THICKNESS, **SEVEN-EIGHTHS** INCH.
Gauged length, 15 inches; cross section, 8.510″ × .885″; area, 7.53 square inches.

Applied loads.		In gauged length.		Remarks.
Total.	Per square inch.	Elongation.	Set.	
Pounds.	*Pounds.*	*Inches.*	*Inches.*	
7,530	1,000	0.0000	0.0000	Initial load.
37,650	5,000	.0018	0.0000	
75,300	10,000	.0041	0.0000	
112,950	15,000	.0066	0.0000	
150,600	20,000	.0090	.0001	
158,130	21,000	.0096	.....	
165,660	22,000	.0101	.....	
173,190	23,000	.0108	.....	
176,000	.....	.0110	.....	
178,000	.....	.0112	.....	
180,000	.....	.0114	.....	
182,000	.....	.0116	.....	
184,000	.....	.0122	.....	
186,009	.....	.0125	.....	
188,000	.....	.0127	.....	
190,000	.....	.0129	.....	
192,000	.....	.0132	.....	
194,000	.....	.0134	.....	
196,000	.....	.0138	.....	
198,000	26,300	.0142	.....	Elastic limit.
200,000	.....	.0150	.....	
202,000	.....	.0156	.....	
204,000	.....	.0164	.....	
206,000	.....	.0191	.....	
208 000	.....	.0920	.....	
210,840	28,000	.1490	.....	
218,370	29,000	.17	.....	
225,900	30,000	.20	.....	
233,430	31,000	.23	.....	
240,960	32,000	.26	.....	
248,490	33,000	.28	.....	
256,020	34,000	.31	.....	
263,550	35,000	.34	.....	
271,080	36,000	.38	.....	
278,610	37,000	.42	.....	
286,140	38,000	.45	.....	
293,670	39,000	.50	.....	
301,200	40,000	.55	.....	
308,730	41,000	.60	.....	
316,260	42,000	.66	.....	
323,790	43,000	.72	.....	
331,320	44,000	.78	.....	
338,850	45,000	.86	.....	
346,380	46,000	.96	.....	
353,910	47,000	1.10	.....	
361,440	48,000	1.23	.....	
368,970	49,000	1.38	.....	
376,500	50,000	1.62	.....	
384,030	51,000	1.92	.....	
391,560	52,000	2.38	.....	
395,100	52,470	2.91	.....	Tensile strength.
395,000	.....	3.25	.....	
392,000	.....	3.65	.....	
0	0	.....	4.98	= 33.2 per cent.

Elongation of inch sections: .18″, .16″, .22″, .27″, .35″, .50″, *1.21″, .50″, .33″, .30″, .25″, .23″, .20″, .17″, 11″.

Area at fracture, 6.59″ × .62″ = 4.09 square inches. Contraction, 45.7 per cent. Appearance of fracture, silky, slightly lamellar.

Fracture began at middle of width of plate. Opening after fracture, .32″; edges closed.

TEST No. 909.—SPECIMEN F_8.—THICKNESS, **SEVEN-EIGHTHS** INCH.
Gauged length, 15 inches; cross section, 8.510″ × .867″; area, 7.378 square inches.

Applied loads.		In gauged length.		Remarks.
Total.	Per square inch.	Elongation.	Set.	
Pounds.	*Pounds.*	*Inches.*	*Inches.*	
7,378	1,000	0.0000	0.000	Initial load.
36,890	5,000	.0020	0.000	
73,780	10,000	.0045	0.000	
110,670	15 000	.0070	0.000	
147,560	20,000	.0097	0.000	
154,930	21,000	.0103	.....	
162,316	22,000	.0108	.....	
169,694	23,000	.0115	.....	
172,000	.....	.0120	.....	
174,000	.....	.0122	.....	
176,000	.....	.0124	.....	
178,000	.....	.0127	.....	
180,000	.....	.0130	.....	
182,000	.....	.0134	.....	
184,000	.....	.0137	.....	
186,000	.....	.0142	.....	
188,000	.....	.0147	.....	
190,000	25 750	.0153	.....	Elastic limit.
192,000	.....	.0165	.....	
194,000	.....	.0182	.....	
196,000	.....	.1775	.....	
206,584	28,000	.23	.....	
213,962	29,000	.26	.....	
221,340	30,000	.28	.....	
228,718	31 000	.32	.....	
236,096	32,000	.35	.....	
243,474	33,000	.38	.....	
250,852	34,000	.43	.....	
258,230	35,000	.47	.....	
265,608	36,000	.52	.....	
272,986	37,000	.57	.....	
280,364	38,000	.63	.....	
287,742	39,000	.69	.....	
295,120	40,000	.74	.....	
302,498	41,000	.84	.....	
309,876	42,000	.92	.....	
317,254	43,000	1.01	.....	
324,632	44,000	1.16	.....	
332,010	45 000	1.28	.....	
339,388	46,000	1.47	.....	
346,766	47,000	1.67	.....	
354,146	48,000	1.98	.....	
361,524	49 000	2.48	.....	
365,700	49.570	3.30	.....	Tensile strength.
0	0	.....	5.31	= 35.4 per cent.

Elongation of inch sections: .17″, .20″, .22″, .25″ .29″ .36″, .49″, *1.41″, .51″, .37″, .29″, .25″, .21″, .17″, .12″.

Area at fracture, 6.43″ × .58″ = 3.73 square inches. Contraction, 49.4 per cent.

Appearance of fracture, silky, lamellar. Fracture open at the middle, .40″; edges closed.

TENSION TESTS OF STRIPS CUT FROM FRACTURED ENDS OF RIVETED JOINTS.

These strips were taken from the middle of the width of the joint plates and parallel to the direction the joints were pulled.

Two strips were taken from each; one was annealed by heating bright red and cooling in the open air; the duplicates were not annealed and were tested as taken from the fractured joint.

When the annealed specimens were at the maximum temperature, center-punch marks, 10″ apart, were stamped on one edge, and about the time the color had left them they were marked again on the other edge. After cooling to 70° Fahr. the distances between these marks were measured.

The amount of contraction, therefore, indicates approximately the heat at which the strips were annealed.

TENSILE TESTS OF STRIPS FROM FRACTURED ENDS OF RIVETED JOINTS.

Joint test number.	Marks on piece.	Contraction in 10 inches from—		Dimensions.		Sectional area.	Elastic limit (approximate).		Tensile strength.		Load at the time of fracture.	
		Bright red.	Black heat.	Width.	Thickness.		Total.	Per sq. inch.	Total.	Per sq. inch.	Total.	Per sq. inch.
		Inch.	Inch.	Inches.	Inches.	Sq. inches.	Pounds.	Pounds.	Pounds.	Pounds.	Pounds.	Pounds.
911	A_2	.104	.081	1.460	.647	.944	31,800	33,690	51,880	54,960	45 800	96,020
		Not annealed.		1.486	.647	.961	34,800	36,210	55.600	57,860	44.900	97,610
913	B_1	.111	.084	1.448	.630	.912	30,600	33.550	52.220	57,260	45 000	97.400
		Not annealed.		1.510	.628	.948	38,000	40,080	56.400	59 490	45 500	89.220
916	D_1	.108	.086	1.440	.844	1.215	37.400	30,780	62,750	51.650	48.000	103.000
		Not annealed.		1.472	.845	1.244	39 500	31,750	66.600	53 540	50.500	95 280
919	E_1	.121	.078	1.473	.850	1.252	41,000	32,750	69,330	55 370	55,500	102,780
		Not annealed.		1.457	.850	1.238	42,500	34 330	70,600	57,030	59 000	91 610
922	G_1	.126	.079	1.460	.847	1.237	40,300	32.580	63 350	51 210	46.000	106,480
		Not annealed.		1.393	.840	1.170	Not well defined.		62.550	53 460	47,800	95,800
925	H_1	.117	.080	1.474	.615	.907	32.700	36,050	52.080	57,420	43,000	101 900
		Not annealed.		1.497	.611	.915	35 000	38,250	54,250	59,290	44 500	98,020
931	K_1	.108	.076	1.450	.610	.885	31,100	35.140	51,040	57.670	41 200	101,970
		Not annealed.		1.408	.605	.852	32.500	38,150	50,800	59,630	41,000	95,790

Joint test number.	Mark on piece.	Condition.	Elongation in 10 inches.		Area of Fracture.	Contraction of area.	Appearance of fracture.	Elongation of inch sections.
			Inches.	Per cent.	Inches. Sq. ins.	P.c.		Inches.
911	A_2	Annealed	2.52	25.2	1.06×.45=.477	49.5	Silky, lamellar.....	.62, .23, .21, .21, .22, .21, .21, .23, .21 . 17
		Not anld.	1.13	11.3	1.07×.43=.460	52.1	do	.37, .17, .12, .09, .08, .06, .07, .05, .06, .06
913	B_1	Annealed	2.40	24.0	1.05×.44=.462	49.3	Silky, stratified.....	.15, .14, .14, .17, .19, .24, .50, .46, .23, .18
		Not anld.	1.33	13.3	1.16×.44=.510	46.2	do	.67, .20, .11, .08, .06, .05, .04, .05, .03, .04
916	D_1	Annealed	2.84	28.4	.95×.49=.466	61.6	Silky, slightly lamellar.	.21, .21, .20, .25, .43, .73, .27, .22, .16, .16
		Not anld.	.82	8.2	1.04×.51=.530	57.4	Silky.....	.06, .03, .04, .05, .05, .05, .09, .07, .14, .24
919	E_1	Annealed	2.97	29.7	1.00×.54=.540	56.9	Silky, slightly lamellar.	.19, .17, .21, .25, .30, .33, .79, .29, .25, .19
		Not anld.	.67	6.7	1.11×.58=.644	48.0	Silky.....	.04, .04, .02, .03, .05, .05, .05, .09, .91, .19
922	G_1	Annealed	3.16	31.6	.90×.48=.432	65.1	do	.16, .16, .19, .22, .28, .31, .32, .90, .34, .28
		Not anld.	.65	6.5	.96×.52=.499	57.4	do	.29, .09, .08, .06, .04, .03, .02, .02, .01, .01
925	H_1	Annealed	2.53	25.3	1.03×.41=.422	53.5	Silky, lamellar.....	.23, .27, .68, .34, .24, .18, .16, .16, .12, .15
		Not anld.	.87	8.7	1.08×.42=.454	50.4	Silky, slightly lamellar.	.36, .15, .12, .09, .05, .03, .02, .03, .01, .01
931	K_1	Annealed	2.32	23.2	1.01×.40=.404	54.4	do	.71, .27, .20, .20, .20, .17, .17, .16, .13, .11
		Not anld.	1.42	14.2	1.02×.42=.428	49.7	Silky, stratified.	.04, .03, .04, .06, .03, .07, .11, .32, .54, .18

TESTS OF RIVETED JOINTS FOR BUREAU OF STEAM

No. of test.	Mark on piece.	Style of joint.	Nominal thickness of plate.	Size and kind of rivets and holes.	Sectional area of plate.	
					Gross.	Net.
			Inch.		*Sq. inches.*	*Sq. inches.*
910	A_1	Double riveted; double butt straps ½ in. thick; 3 in. pitch.	⅝	¾ inch steel rivets; $\frac{25}{32}$ inch drilled holes.	13.22	10.12
911	A_2		⅝		13.08	10.01
912	A_3		⅝		12.41	9.50
913	B_1	Treble riveted; double butt straps ½ in. thick; $3\frac{9}{16}$ in. pitch.	⅝	¾ inch steel rivets; $\frac{25}{32}$ inch drilled holes.	12.81	10.31
914	B_2		⅝		12.69	10.21
915	B_3		⅝		12.691	10.21
916	D_1	Double riveted; double butt straps ¾ in. thick; 3¾ in. pitch.	⅞	1 inch steel rivets; $1\frac{1}{16}$ inch drilled holes.	14.671	10.09
917	D_2		⅞		14.646	10.07
918	D_3		⅞		14.705	10.11
919	E_1	Treble riveted; double butt straps ¾ in. thick; $4\frac{9}{16}$ in. pitch.	⅞	1 inch steel rivets; $1\frac{1}{16}$ inch drilled holes.	14.322	10.63
920	E_2		⅞		14.438	10.72
921	E_3		⅞		14.256	10.58
922	G_1	Leavitt joint; double butt straps; one of usual width for double riveting and ⅞ inch thick; other, ⅝ inch thick and extended far enough on each side to receive five additional rivets in two rows. (See Plate.) Pitch of double riveting, 2⅝ inch.	⅞	3 plates; $1\frac{1}{16}$ inch iron rivets; 1⅛ in. drilled holes.	13.781	10.93
923	G_2		⅞	2 plates; 1⅛ in. iron rivets; 1¼ inch drilled holes.	13.852	10.985
924	G_3		⅞		13.741	10.90
925	H_1	Leavitt joint; same arrangement as in G series except that wide butt strap has six rivets on each side beyond narrow strap. Pitch of double riveting, 2⅞ inch.	⅝	3 plates; 1 inch inch iron rivets; $1\frac{1}{16}$ in. drilled holes. 2 plates; 1⅛ inch iron rivets; 1¼ inch drilled holes.	13.252	10.537
926	H_2		⅝		12.776	10.154
927	H_3		⅝		12.81	10.186
928	I_1	Leavitt joint; same arrangement as in G series except that the five rivets in ends of wide butt strap are differently spaced. Pitch of double riveting, 2⅞ inch.	$\frac{9}{16}$	$1\frac{3}{16}$ inch iron rivets; 1 inch drilled holes.	8.287	6.820
929	I_2		$\frac{9}{16}$		8.211	6.755
930	I_3		$\frac{9}{16}$		8.28	6.81
931	K_1	Treble riveted; double butt straps ½ in. thick; $3\frac{9}{16}$ in. pitch.	⅝	¾ inch iron rivets; $\frac{25}{32}$ inch drilled holes.	12.36	9.94
932	K_2		⅝		12.93	10.40
933	K_3		⅝		12.98	10.44

* No figures given because no tests were made of this thickness of metal for tensile strength.

ENGINEERING, UNITED STATES NAVY DEPARTMENT.

Bearing surface of rivets.	Shearing area of rivets.	Tensile strength of plate per sq. inch.	Maximum stress on joint per square inch.					
			Tension on gross section of plate.	Tension on net section of plate.	C'mpress'n on bearing surface of rivets.	Shearing on rivets.	Efficiency of joint.	
Sq. inches	*Sq. inches*	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*	*Per cent.*	
6.72	12.46	53 710	42.860	55 990	84.320	**45,470**	79.8	a
6.64	12.46	53.710	40,960	**53,530**	80,690	43.000	76.2	b
6.30	12.46	53 710	42.720	**55,800**	84.140	**42,540**	79.5	c
8.01	15.34	53 710	43.460	**54,040**	69.560	36.320	80.9	d
7.93	15.34	53.710	40.390	**50,200**	64 630	33.410	75.2	e
7.94	15.34	53 710	44.290	**55,050**	70,790	36,640	82.5	f
8.25	15.96	51,190	35.180	**51,150**	62.560	32,340	68.7	g
8.23	15 96	51,190	36.190	**52,640**	64 410	33,210	70.7	h
8.27	15.96	51,190	35.780	**52,050**	63.630	32.970	69.9	i
10.15	19.51	51,190	37.910	**51,080**	53.500	27,830	74.1	j
10.23	19.51	51,190	38.400	**51,720**	54.190	28.420	75.0	k
10.10	19.51	51,190	37.950	**51,130**	53.560	27.730	74.1	l
16.30	28.00	51.190	40,810	**51,460**	34.500	20,090	79.7	m
16.38	28.00	51,190	41.740	**52,640**	35.300	20,650	81.5	n
16.26	28.00	51,190	40,120	**50,580**	33.910	19,690	78.4	o
14.045	30.41	53.710	42,820	**53,860**	40,410	18,660	79.7	p
13.548	30.41	53.710	45.300	**57,000**	42.720	19.030	84.3	q
13.576	30.41	53.710	46,070	**57,940**	43,470	19,410	85.8	r
8.057	18.06	*	42,250	**51,330**	43.450	19,380	*	s
7.994	18.06	*	40.800	**49,590**	41.910	18,550	*	t
8.05	18.06	*	40,720	**49,520**	41,890	18,670	*	u
7.773	15.34	53.710	43.600	**54,220**	69,720	**35,130**	81.2	v
8.08	15.34	53 710	43 260	**53,780**	69.220	36.460	80.5	w
8.11	15.34	53.710	43,000	53,460	68,820	**36,390**	80.1	x

Figures in heavy-faced type indicate manner of failure.
For explanation of reference letters in last column, see next page.

MODE OF FRACTURE AND APPEARANCE OF FRACTURED SURFACES.

a Sheared the rivets in one plane in Plate A; started a fracture at side of one rivet hole in outside row of riveting.

b Fractured Plate A along outside row of rivet holes. Appearance of fractures, silky, lamellar.

c Fractured Plate A along outside row of rivet holes; sheared (double shear) six rivets in Plate B. Appearance of fractures, silky, slight lamination.

d Fractured Plate A along outside row of rivet holes. Appearance of fractures, silky.

e Fractured Plate A along outside row of rivet holes. Appearance of fractures, silky, slightly lamellar.

f Fractured Plate A along outside row of rivet holes. Appearance of fractures, silky, lamellar.

g Fractured Plate B along outside row of rivet holes; tore apart from one edge. Fractures also started in Plate A at opposite edge. Appearance of fractures, silky, lamellar.

h Fractured both plates along outside row of rivet holes. The separation of Plate A was complete; Plate B fractured through four sections. Appearance of fractures, silky, slightly lamellar.

i Fractured both plates along outside row of rivet holes. Plate B did not separate at one edge. Appearance of fractures, silky, slightly lamellar; metal well drawn down.

j Fractured Plate B, taking zig zag course through two outside rows of rivet holes. Appearance of fractures, silky in part, granular in part; the metal in the silky sections well drawn down, the granular sections not much reduced in thickness, the extremes of thickness after fracture being .665″ in the silky and .840″ in the granular metal. A loud report accompanied the fracture of the granular metal.

k Fractured Plate A, taking a zig zag course through two outer rows of rivet holes. Appearance of fracture, silky, slightly lamellar.

l Fractured Plate B, taking a zig zag course through two outside rows of rivet holes. Fracture silky, slightly lamellar.

m Fractured Plate A along outside row of rivet holes. Fracture silky, slightly lamellar. Mean thickness at fracture, .56 inch.

n Fractured Plate A along outside row of rivet holes. Fracture, silky, slightly lamellar.

o Fractured Plate A along outside row of rivet holes. Fracture, silky, lamellar. One seam in fractured surface ¾″ wide.

p Fractured Plate A along outside row of rivet holes. Fracture, silky, lamellar.

q Fracture in same place as H_1. Appearance, silky, slightly lamellar.

r Fractured Plate B along outside row of rivet holes. Appearance, silky, slightly lamellar. Plates open at butt joint ⅛ inch.

s Fractured Plate B along outside row of rivets, beginning the fractures at edges and extending from rivet holes toward middle of plate. Fracture, silky, slightly lamellar; metal well drawn down.

t Fractured Plate A along outside row of rivets. Appearance of fracture, silky, slight lamination.

u Fractured Plate A along outside row of rivets. Appearance of fracture, silky, slightly lamellar; metal well drawn down.

v Fractured Plate B; followed outside row of rivet holes in part, and thence, through inside rows, to end of plate; sheared two end rivets. Fractures, silky, slightly lamellar.

w Fractured Plate A along outside row of rivet holes, except end sections and one middle section. Appearance, silky, lamellar.

x Sheared every rivet in the joint in both plates. The under butt strap dropped to the floor. Double shear in Plate A, three rivets; single shear in Plate B, with the exception of one rivet, which sheared in two planes.

COAL ENDURANCE AND MACHINERY OF THE NEW CRUISERS.

A Course of Lectures Delivered at the Naval War College, Newport, R. I., in October, 1892.

By Passed Assistant Engineer Ira N. Hollis, U. S. Navy.

INTRODUCTION.

The following lectures on the coal endurance of the new cruisers and the care of machinery were prepared for the Naval War College, and are now published, with some revision and additions, in the Engineers' Journal. Many unavoidable inaccuracies have crept into the tables through lack of information on the exact condition of steaming and the quality of coal; but it is hoped that the results will be found sufficiently correct for all practical purposes. I have not tried the kindergarten system, as the types of engines and boilers, together with the names of their parts, are generally well known by all officers of the Service. Besides, the possession of such information by the commanding officer of a ship does not show any real knowledge of the subject. The questions on steam machinery given to line officers in their examinations for promotion do not seem to me to fill the requirements of the case in our new ships. I have therefore endeavored to state briefly those things appertaining to steam machinery which I think a commading officer ought to know. All questions of mechanical details must be left to the engineer, whose judgment on such subjects will be better than that of the deck officer, however much the latter may have read.

There is no intention to reflect upon the work of any department or corps, but rather to state the exact truth in regard to our ships without partisanship, so far as I have been able to see it. Nobody will be happier than I to be set right upon questions open to controversy.

LECTURE FIRST.

Steaming Qualities of the New Cruisers.

Many of the class of 1878 will recall the subject of an essay given out at the Naval Academy by Professor Soley, now the Assistant Secretary of the Navy, "The Warships of the Future." It would be interesting to know how many youthful predictions have been realized in our new cruisers and modern battleships. Few were rash enough at that time to describe a ship without sail power and dependent wholly upon her coal. The change of design has been so startling that our officers have not yet adjusted themselves to this new condition, and few understand the elements which go to make the modern cruiser a success. It has given rise to an entirely different set of questions. Formerly, no one thought of the endurance of a ship and the economical rate of steaming; now, it is the all important question. The handling of guns is easier, the navigation is the same, and the actual management of the engines under steam is quicker with the separate air and circulating pumps, and the making of steam far more rapid with the great air ducts and blowers. The workmanship on machinery has improved so much that the actual running of main engines, starting and stopping, are wonderfully simple, but the care and judgment required to keep the engines and boilers, especially auxiliaries, up to their highest capacity are enormously increased. As most officers know, the change from the old navy is mainly in the care and preservation of machinery and boilers, the coal supply and the prevention of corrosion and fouling.

COAL ENDURANCE.

The first subject to be taken up is the endurance of the new ships as deduced from the log books, and I will make it as practical as possible. The theory of speed curves, the measured-mile trials and preliminary calculations, are well enough in their way, only as guides in the design of new ships, but they are almost valueless to the officer who has actually to command the ship or to look after the coal account. I have in mind the story of a scientific man who pointed out a lump of coal to a student

and asked him what he saw there. He saw only a piece of coal, but the professor saw the whole carboniferous age. He would not have made a good fireman, nor do I think that the vision of a graceful speed curve, which contains within itself all the laws of hydraulics, would get the captain of a man-of-war from port to port, especially if he got out of coal. What he wants to know before leaving port is of a nature entirely practical, what speed can be made, what coal required, and how much left in the coal bunkers upon arriving at the destination. It is the information that any officer may obtain with a small note book and pencil. The data necessary to form correct conclusions in regard to our new ships now in commission I hope to supply, not from the preliminary calculations made by the Bureaux before the ships were built, but from the steam logs.

I take the case of the *Charleston* first, as she has steamed over forty thousand miles in two years, during which time I was attached to her. The effect of adverse weather, bad coal and fouling of the bottom will all be shown in the subjoined table. Her theoretical steaming radius, as given in the report of the Chief of Bureau of Construction for 1887, is as follows:

Table I.

U. S. S. *CHARLESTON.*

ENDURANCE FROM THEORETICAL CALCULATIONS.

I. H. P.	Speed.	Coal per day	Endurance on 328 Tons coal.	Endurance on 760 Tons coal.	Coal per I. H. P.	Endurance. in service.
7,650	18.9	180	805	1,906	2.2	
6,000	17.5	135	995	2,367	2.1	
5,000	16.5	107	1,183	2,811	2.0	
2,750	13.0	59	1,697	4,033	2.0	
1,200	10.0	26	2,990	7,103	2.0	(2920 Foul. (4450 Clean.

Table No. 2 containing the economic qualities of the ship is made out from the records for all the long voyages of the cruise. The coal used from port to port, including all expenditures, and, in most cases, even that for getting up steam, is simply divided by the number of days in the voyage, and the endurance worked

Table II.

U. S. S. *CHARLESTON.*

DISPLACEMENT = 3,730 TONS. MAXIMUM SPEED = 18.21. BUNKERS = 760 TONS ANTHRACITE.

STEAMING RADIUS AND ENDURANCE FROM THE STEAM LOG.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
No.	Knots speed per hour.	Revolutions per minute.	Revolutions per knot per minute.	Revolutions per knot.	Knots per day.	Tons coal per day.	Knots per ton of coal.	Days steaming.	Knots' steaming radius.	Coal. Kind.	Coal. Quality.	General direction of wind.	Kind of sea.	Months out of dock.	Condition of bottom.	No. days in voyage taken.	No. boilers used.
1	6.8	50.9	7.49	449	163 0	35 3	4.62	21 5	3509	B	Good	Var.—light	Mod.	5	Foul	15	3
a 2	8 8	57 3	6.51	391	210.5	53.4	3 94	14.2	2995	B	Bad	Var.—light	Smooth	6	Foul	11	3⅓
b 3	10.0	65.3	6.53	392	246.5	62.1	3 97	12.2	3007	A&B	Good	Var.—light	Var.	8	Foul	9	4½
c 4	8 3	50.5	6.09	365	199.3	31.6	6 31	24 05	4793	B	Good	Beam—var.	Heavy B'm	3	Clean	17	3
5	8.7	49.4	5.68	341	208.3	33.1	6.29	22.90	4781	B	Good	Light	Smooth	1	Clean	17	3
6	8 8	51.6	5.86	352	211.0	30.3	6.96	25.09	5293	B	Good	Light	Smooth	2	Clean	20	3
7	9.2	54.0	5.87	352	221.0	32.2	6.86	23.61	5216	B	Best	Light	Smooth	1	Clean	4	3
8	10.2	59.2	5.80	348	245.0	44.0	5.57	17.28	4233	B	Good	Light	Smooth	½	Clean	9	4
9	11.0	64.5	5.86	352	262.6	60.8	4.32	12.50	3282	A&B	Bad	Light	Smooth	1	Clean	9	4
10	13.8	86.7	6.28	377	331.3	89.0	3.72	8.54	2828	B	Bad	Variable	Var.	½	Clean	6	6
11	7.4	49.0	6.62	397	178.7	36.7	4.87	20.71	3701	B	Good	Head—5-9	Heavy H'd	4	Clean	10	3
d 12	9.9	59 1	5.97	358	236.4	46.3	5.11	16.40	3879	B	V'y b'd	Var.—light	Smooth	1½	Clean	9	3

a Screw and after run scraped. *b* Bottom scraped by divers. *c* Very rough sea. *d* Heavy head wind and sea.

Steaming radius with only absolutely necessary expenditure for auxiliary purposes.—Knots speed per hour, 8.8; knots per day, 211; tons coal per day, 28.3; knots per ton of coal, 7 46; knots steaming radius, 5,669.

Expenditures for galley, dynamo, distilling, &c., per day = 2.9 tons.

Economical revolutions = 51 per min. Speed = 8 75 knots with clean bottom.

The longest run made by the ship was in June, 1891, from Iquique to San Diego, 4.308 knots by the log. She burned 619 tons of West Hartley coal, and made the voyage in 20 days, 10 hours, under half boiler power, natural draft, and had over a hundred tons in the bunkers upon arriving in port. The rate of speed was 8 8 knots on 30.3 tons of coal. Weather favorable and bottom clean.

NOTE.—In the tables throughout this lecture, the letters A and B, in column 11, stand for anthracite and bituminous respectively.

out from the result. The revolutions per knot are supplied in order to show any undue resistance in the propulsion of the ship. It will be noted that the average was about 350 revolutions for one knot, or 5.83 revolutions per minute per knot at ordinary speeds, and that it rose to 449 with a very foul bottom and 397 against a strong head wind.

The first three runs contain the performance with a foul bottom after having lain in Honolulu several months. Run No. 1 was made without attempting to scrape the bottom, No. 2 after having scraped the screw, and No. 3 after the divers had worked several weeks on the bottom. The economical speed is 6.8 in the foul condition. The next seven runs are made with fairly clean bottom and the economical speed rises to 8.8. With very strong head wind, given in No. 11, the speed was 7.4.

In all these cases, however, the revolutions were about the same—51 per minute—and this can safely be taken as the most ecomomical rate, without regard to weather, sea or bottom. A following or beam sea and wind did not affect the ship much. I have also added a theoretical run at 8.8 knots, supposing the dynamo shut down, only enough water to drink distilled, and no steam used for deck purposes. The bunker capacity is given at 760 tons, although that is, strictly speaking, the contents for anthracite coal, but enough bituminous coal can always be carried on deck to bring the weight up to that.

Table III.

U. S. S. *YORKTOWN.*

ENDURANCE FROM THEORETICAL CALCULATIONS.

Speed.	I. H. P.	Tons coal per day.	Distance per day.	Steaming radius on 380 tons	Days.	Coal per I. H. P. per hour.	Steaming radius, actual service.
16	3,300	61.7	384	2,338	6.1	1.75	
15	2,620	42.	360	3,254	9.04	1.50	
14	2,000	31.92	336	3,998	11.90	1.50	
13	1,600	25.68	312	4,614	14.80	1.50	
12	1,230	19.70	288	5,558	19.30	1.50	
10	650	11.04	240	8,258	34.41	1.60	4,500
8	375	6.24	192	11,689	60.68	1.80	4,800
6	200	4.08	144	13,408	93.11	2.00	

Table IV.

U. S. S. *YORKTOWN.*

DISPLACEMENT = 1700 TONS. MAXIMUM SPEED = 16.65 KNOTS. BUNKERS = 380 TONS ANTHRACITE.

STEAMING RADIUS AND ENDURANCE, FROM STEAM LOG.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
No.	Knots speed per hour.	Revolutions per minute.	Revolutions per knot per minute.	Revolutions per knot.	Knots per day.	Tons coal per day.	Knots per ton of coal.	Days steaming.	Knots steaming radius.	Coal. Kind.	Coal. Quality.	General direction of wind.	Kind of sea.	Months out of dock.	Condition of bottom.	No. days in voyage taken.	No. boilers used.
a 1	7.8	79.3	10.2	610	186.7	22.6	8.26	16.8	3140	A	Good	Beam—strong	Rough	3	Rough	10	3
b 2	8.8	82.4	9.4	562	210.7	20.1	10.48	18.9	3986	A	Good	Var.	Smooth	½	Clean	9	3
c 3	9.2	89.8	9.8	586	221.9	29.3	7.57	13.0	2878	A	Good	Ahead—light	Smooth	3½	Rough	6	4
d 4	8.4	84.5	10.1	604	202.5	16.0	12.66	23.8	4811	B	Good	Var.—light	Smooth	7	Rough	21	2
e 5	10.0	96.6	9.7	580	239.4	22.5	10.64	16.9	4043	B	Good	Ahead—light	S.—m.	2¾	Clean	19	3
f 6	11.1	108.0	9.7	584	266.0	32.8	8.11	11.8	3081	B	Good	Var.	S.—m.	4½	Rough	10	4
g 7	9.6	87.4	9.1	546	230.5	19.3	11.94	19.7	4538	B	Best	Beam—mod.	Mod.	1	Clean	5	3
h 8	10.4	92.9	8.9	536	250.0	21.3	11.74	17.8	4461	B	Best	Var.—mod.	Mod.	1½	Clean	6	3
i 9	11.6	109.0	9.4	564	277.5	28.6	9.70	13.3	3686	B	Best	Var.	Smooth	3	Rough	14	3½
j 10	7.1	73.3	10.3	619	170.0	16.0	10.62	23.8	3936	B	Good	Beam—strong	Mod.	2½	Rough	4	2½
k 11	9.4	92.8	9.9	592	226.4	22.6	10.20	16.9	3876	B	Bad	Bow—strong	Mod.	2	Clean	6	3

a Fore-and-aft sails, 5 days. *b* Fore-and-aft sails, 2½ days. *c* No sails. *d* Gale on beam 2 days; no sails. *e* Fore-and-aft sails, 1 day.
f Fore-and-aft sails, 1½ days. *g* Fore-and-aft sails, 4 days. *h* No sails. *i* Fore-and-aft sails, 8 days. *j* No sails. *k* No sails.

Steaming radius, with only absolutely necessary expenditures for auxilaries.—Knots speed per hour, 8.4; knots per day, 202.5; tons coal per day, 15.0; knots per ton of coal, 13.50; knots steaming radius, 5130.

Expenditure for galley, dynamo, distilling, &c., per day = 1.5 tons.

Economical revolutions = 85 per min. Speed = 8.8 knots, with clean bottom.

The longest run made by the ship was in March, 1892, from Callao to San Francisco, a distance of 4191 knots by the log, which she made in 20 days, 17 hours, at the rate of 8.4 knots, under half boiler power, natural draft. The total amount of coal used was 331 tons of good bituminous. The weather was generally favorable and the bottom somewhat foul. No sails.

Table No. 3 gives the endurance and steaming radius of the *Yorktown* calculated theoretically. Table No. 4 is worked up from the log. It explains itself, and I will only call attention to the great falling off when using anthracite. The effect of fore-and-aft sail in increasing the steaming radius cannot be determined, as all runs without sails were made under unfavorable conditions. The results show some discrepancies which the sea and sail probably account for. The economical rate seems to be at 85 revolutions, giving nearly nine knots with clean bottom.

In this connection we have two reports from different commanding officers of the ship, showing conclusively how deceptive short runs may be. In October, 1890, her commander stated, as a result of twelve hours' steaming in Long Island Sound under favorable circumstances, that the *Yorktown* would steam 200 miles a day on 12 tons of coal. (She was probably at normal draught, in smooth water, with clean bottom and good coal.) Commander Evans reports that after several trials of the ship at sea the most economical rate of steaming was found to be 200 miles per day on 17 tons of coal, a result fully borne out by the log book for several voyages. She is put down at 93 days' endurance in the theoretical calculations, whereas she could not keep the sea above 25 days with any economy except under banked fires.

The tables for the other ships explain themselves. A comparison will establish the fact that the *San Francisco* is the most economical of the new cruisers and the *Philadelphia* has the greatest steaming radius. Both of them are well ahead of the *Charleston* and *Baltimore*, which are both of English design.

To obtain the best results on a long passage, the revolutions should be set and enough coal burned to run the engines at that rate. This eliminates all questions of displacement, weather and foulness of bottom. It also removes the temptation to steal coal, which is almost irresistible to a fireman with low fires.

I have added a theoretical run with every ship, to show the greatest possible endurance in case of an emergency, when it becomes necessary to shut off all the deck auxiliaries.

All the tables show the rapid decrease of speed from foulness

of bottom and consequent increase of expense. Take the case of the *Charleston's* voyage from Honolulu to the Coast. She burned 460 tons on a run of 2,130 knots in 13.03 days, and only 419 tons for the same distance on a return voyage in 9 days with a clean bottom. Had the former run been made in 9 days the coal expenditure would have been about 765 tons, an increase of 346 tons. This would have added $5,000 to the running expenses, if the ship could have carried enough coal to make the distance. It also shows to what extent the steaming radius dwindles away.

The screw was thoroughly scraped in Seattle and a small part of the after run, and the ship showed an immediate gain of speed at the same number of revolutions. In ordinary cruising it is an easy matter to keep the screw bright, even if the hull cannot be scraped by divers. Its surfaces move through the water at so high a rate of speed that very little deposit will affect the coal consumption materially. A simple calculation in the case of the *Charleston* will show this very well. When the ship is making 10 knots the tips of the blades are making 27.3 knots, and one square foot of the screw surface is equivalent to about 11.25 square feet of the hull. Each screw has a blade surface of 54.75 square feet, so that both screws, taking back and face, are equivalent to 2,464 square feet of the wetted surface at 10 knots, or one eighth of the whole hull.

In general terms, the economical speed of our ships will be found from one-half the maximum speed to one knot above it, varying somewhat with the coal expended for auxiliary purposes. The coal consumption is in the neighborhood of one ton for 100 tons of displacement for ships below 5,000 tons. These results can be obtained in most cases with one-half boiler power natural draft.

SUSTAINED SEA SPEED.

The sustained sea speed can be stated at about four-fifths of the maximum speed for four hours on forced draft.

The *Charleston* has the longest run at full power of the new ships. She started from San Francisco on two-thirds power, natural draft, added the other boilers in twelve hours and steamed

full power with ½-inch pressure in the fire room and the blowers running half speed for four days, when a drain cock blew out on one of the boilers and spoiled the record for the voyage to Honolulu. During this period, the distance made by the log for forty-eight hours, in a moderate sea with a light wind, was 707.4 knots, an average of 14.74 per hour. The distance by observation was 726, or 15.13 per hour. It may be fairly assumed that she could make 14.75 knots with clean bottom under ordinary circumstances of wind and weather.

The *Baltimore* has a record of 697.4 knots for forty-eight hours on the voyage from Copenhagen to Lisbon, an average of 14.53 per hour, using full-power natural draft. She could probably have brought this up a knot with the blowers running at low speed.

The *Yorktown* made 576.4 knots for 48 hours, 12 knots per hour, on the run from Porto Grande to New York, with full-power natural draft. This might also have been increased a knot by running the blowers.

The *Philadelphia* ran 611 knots in 48 hours, 12.75 knots an hour, with all boilers, natural draft, using mixed coal. This is no criterion of her performance, as she was probably not run up to her speed. The above examples are taken from long runs after the fires had become dirty and the water salt.

Mr. White says, "The English Admiralty practice is based on a rough rule that the maximum power developed at sea over long periods will not be more than two-thirds, or less than one-half, the natural-draft power developed on the contractor's trial." The trial of 21 ships at sea for periods of about 96 hours at full-power natural draft gave 86 per cent. of the power developed on the contractor's trial, but the ships varied from 71 to 102 per cent., thus proving that the contract trial does not establish much for the cruising speed of any given ship. The following estimate of the cruising speed for two ships is based upon the contract trials:

H. M. S. *HOWE*.

		Knots speed.
Contractor's Trials—Forced Draft—	11,600 I.H.P.—	16.9
" " —Natural " —	8,200 " —	15.9
Continuous steaming (estimated) —	4,500 " —	13.5

H. M. S. *MEDUSA.*

			Knots speed.
Progressive Trial—Forced	Draft—10,000	I.H.P.—	19.9
" " —Natural	" — 6,300	" —	18.0
Continuous steaming (estimated)	— 3,500	" —	15.75

The above rule of four-fifths the maximum speed is sufficiently accurate for all purposes and is much simpler.

In giving out the contract for a fast cruiser it would seem to be far better to require a maintained sea speed under reasonable conditions specified in the agreement, and, in addition, to require a maximum power trial under forced draft for four hours to test the strength of engines, without regard to speed.

In the use of forced draft for chasing or manœuvering, the limit is reached in about six hours for running the blowers full speed. This depends upon the system of draft and the design of boilers. The closed ash pit would probably be good for a much greater period. During the *Charleston's* trial off the coast of California, we found the utmost limit to be ten hours. The blowers were run that long and the tube ends became so clogged up with salt deposit that half of them were out of commission. The combustion chambers were also a foot deep with ashes and soot.

TRIALS WITH ONE SCREW.

One of the great advantages of twin screw ships arises from duplicate parts and immunity from detention in case of breakdown at sea. The following ships have been tried with one screw during repairs to the other engine: On September 1, 1890, the *Charleston* ran the starboard engine at 68 revolutions for 4 hours, and made 7.5 knots per hour, or about two thirds of the speed with two screws. The allowance of coal was not changed. On September 8, 1890, the *Baltimore's* port steam-pipe blew out, and the starboard engine ran 13 hours alone at 70 revolutions. She made 9.6 knots per hour, about three-fourths of the speeed with two screws. The *San Francisco* steamed 8 hours with the starboard engine, making 59.5 revolutions and 8.05 knots, about four fifths of the speed with both screws.

The speed obtained with one screw would probably be about

three fourths of that obtained with both on the same consumption of coal. This applies to all the ships with twin screws.

TIME REQUIRED FOR INCREASING SPEED.

It may be useful to a commanding officer to know the time required to bring the ship up in speed when boilers have to be added. With blowers ready for use, a very short time will suffice to gain one or two knots, but when other boilers have to be added, two or three hours will be necessary. I cannot do better than to go back to the *Charleston* for data, taken from her log during the chase of the *Itata*.

A strange ship had been sighted on May 30 well out to sea, and, on account of her suspicious behavior, Captain Remey decided to overhaul her. At 9 A. M. the *Charleston* was making 60 revolutions and about 10.9 knots with steam and fires in three after boilers, natural draft. The forward boilers were empty and the grates bare. By starting one after blower (the other being out of commission) the speed was brought up to 12.2 knots immediately. The forward boilers were run up and the fires started in the center boiler at 9.15 A. M. Steam formed at 10.40, and the connecting valve was opened at 11.30; fires started in the port boiler at 10.30 gave steam to the engines at 12.45. The forward blowers were started at noon. The above two boilers brought the speed up to 13.5 knots by 12.30, a gain of only 1.3 knots over the after boilers. It required 2 hours 15 minutes to get steam from cold boilers, but 3½ hours to get the boilers into good steaming condition from the time the order was given. Had the boilers been run up, coal on the grates and the water heated to boiling point by the hydrokineters, the time of getting steam for effective work could have been reduced to less than one hour.

This is very suggestive as to the best method of keeping a ship in proper condition for emergency. The spare boilers should all be at steaming level, with furnaces primed and water kept at boiling point by the hydrokineters. There would be no necessity of banked fires, as the boilers would already have steam formed. Also, lying in port where there may be a sudden call

for the ship, it will be sufficient to have steam and fires in one boiler and the water kept hot in the others by the hydrokineters with the fires ready to light. The time required to start the auxiliaries and get the main engines ready to run will give ample time to get the boilers connected. The above is all in the line of saving coal, not by vexatious allowances during steaming, but by judicious expenditures in port. It also lightens the firemen's work. Upon arriving in port, allow fires to die out in all boilers except one, and clean furnace and tubes thoroughly, while the water is kept hot by steam from the boiler in use.

REASONS FOR THE FALLING OFF IN ENDURANCE.

Much of the unfavorable comment passed upon our ships when compared with the English and French arises from a false method of comparison. We know only the preliminary calculations of their ships, which are probably as accurate in practice as our own. The falling off is no cause for discouragement, however, as the new ships have fair speed and can get somewhere when they start. The *Charleston* was ordered on May 8th to go south in chase of the *Itata*. She went to Iquique, repaired the latter ship and was back in San Diego by July 4. An old time vessel would have been out in the Pacific looking for a breeze or in Panama taking coal.

There are many reasons for this great discrepancy between the ideal ship with unlimited endurance and the best of the above cruisers whose utmost limit is 7,000 knots. The most important among them are as follows:

1st. The calculations are made at normal draught and displacement while the average draught is always greater in actual steaming. It is usually found necessary to fill several compartments of the double bottom after using the coal from the protective deck; so that the ship is not far from her normal draught upon arriving in port, thus making the average for the voyage much greater. Take the *Charleston*, for instance. She would leave port with full bunkers, drawing 19 inches more water than with normal load, and arrive at her destination, if the voyage had been long, drawing about 4 inches less, thus giving an

average mean draught for the voyage 7½ inches above the normal. This represents 173 tons displacement. A better example occurred in the voyage of the *Concord* from Montevideo to Barbados. She left port with 440 tons of coal and mean draught of 15 feet 1½ inches, and arrived, after 18 days 18 hours at sea, with 17 tons of coal and 13 feet 6 inches mean draught, making an average mean draught for the voyage of 14 feet 3¾ inches, just 3¾ inches deeper than the normal draught, with 51.5 tons greater displacement.

2d. The trim is rarely that for which the ship is designed. Experience proves that any departure from it is generally a cause of loss. As the boilers are placed in at least two separate water tight compartments, one forward of the other, the coal has to be taken out either forward or aft. It is impracticable to divide the engineer's force and carry steam in one boiler of each fire room; besides, when a ship has much steaming to do, the best opportunities for cleaning boilers occur at sea, and it is therefore desirable to keep one fire room clear. The plan followed on the *Charleston* seems a very good one—to use the forward boilers until the saturation got up to 4, then shift to the after boilers, and put a working gang into the forward fire room for cleaning. In this way, from two to three hundred tons of coal would be used out forward, bringing the ship well up at the bow, followed by a decrease of speed. The experiment of filling the double bottoms forward after using 200 tons of coal was tried several times with a decided gain in speed for the same consumption of coal, probably due to bringing the ship nearer to her lines. No allowance can be made for this loss of efficiency in a preliminary calculation, as it must vary greatly in different ships. Any list would also decrease efficiency.

3d. The ship's bottom is always more or less foul. A very frugal collection of barnacles will decrease the speed greatly. I cannot do better than to cite the case of the *Charleston* again. With clean bottom and good coal she made about 6.9 knots for one ton of coal at 9 knots speed. (The theoretical calculation gives 9.23 knots on one ton at 10 knots speed.) After lying in Honolulu three months without sending the divers down, she

made on a voyage of 15 days just 4.62 knots per ton at 6.8 knots speed, using Wellington coal, the same as above. She returned to Honolulu in one month and lay two months in port, during which period the divers were sent down to scrape the bottom thoroughly. Her next voyage, 9 days, gave 3.84 knots for one ton of coal at 10 knots speed, using anthracite and Wellington coal.

4th. The sea and weather also exercise a depressing influence upon the endurance. The wind and sea on the beam or astern do not help the ship much unless she has sails, but a head wind and sea knock down the speed very rapidly. The *Charleston* started from Honolulu for Acapulco on the seventeenth of last December and had the wind dead ahead for about 9 days with a force varying from 6 to 9. The head sea was also very heavy. Her performance during this period was 4.87 knots on one ton of coal at 7.4 knots speed. On the other hand, with sails the wind may be a great help; but a following sea is usually bad on account of racing. In this case the size of the ship has much to do with the effect of sea and weather. The larger and longer the less the effect. The enormous ocean racers make their trips without much attention to adverse weather.

5th. The first calculations are based on trial trips, where the best of coal is used; whereas, in service such coal is rarely obtained. The enormous loss due to this cause alone is well shown by the *Charleston's* cruise. At Seattle she took black diamond coal and obtained 57.3 revolutions per minute on 53.4 tons of coal per day, while she made on another trip to Honolulu 59.2 revolutions with 44 tons of Wellington coal per day. I cite the revolutions, because the condition of the bottom was not the same in both cases.

6th. No allowance is made for distilling, dynamo and galley. This varies from 1 ton a day in the small ships to 3 or 4 in the moderate-sized cruisers. I base the amounts given on the records of the log book. This subject will be referred to in another place.

7th. The horse power required to drive the ship is usually much higher than that given on the speed curve or in the pre-

liminary calculations. Take the case of the *Concord* again on her voyage from Montevideo to Barbados; she averaged 9.4 knots and 86.1 revolutions the entire voyage, with her fore-and-aft sails set about half the time. The average horse power taken from 18 sets of cards was 832.9. The measured-mile trial of the *Yorktown*, an exactly similar ship, gave 9.4 knots on 530 I.H.P., a difference of 302.9. Most, if not all, of this is due to the causes cited above.

8th. Some of the loss is undoubtedly chargeable to bad firing. This element is often great, particularly if a new ship is pushed to high speed with a green crew. Its amount cannot be estimated, but that the chief reduction in speed is strictly due to deeper draught, foul bottom, unfavorable weather and other conditions outside of the fire room, is shown conclusively in the *Concord*, where a horse power of 832.9 was obtained and the ship made only 9.4 knots.

9th. The size has a great influence upon efficiency. In maintaining speed at sea, the heavier and longer ship has the advantage for a given speed, both as to weight of machinery in proportion to the hull and the influence of sea and weather. As Mr. White says, the small ship seems to feel the check of the sea more than a large ship. I have no doubt that many officers have observed this. An excellent illustration is furnished in the trials of the English ships, where the *Dreadnought* made 61½ knots in four hours on full power at sea. Her displacement was 10,600 tons, and the above speed of 15.38 knots was obtained on 76 I.H.P. for every 100 tons of displacement. The *Scout* made 58 knots in the same time on full power with a displacement of 1,600 tons. The speed was thus 14.5 knots on 202 I.H.P. for 100 tons of displacement. We actually have the case of a slower ship being able to overtake a fast ship at sea that she could not touch in smooth water.

Mr. White, in the last volume of the Transactions of the Institute of Naval Architects, gives some interesting data about recent ships. He says:

"To members of this institution it is unnecessary to explain the causes which tend to economy of power with increase in di-

mensions, but the figures have an interest in connection with warship design, and they illustrate the difficulties to be faced in obtaining high speeds in cruisers and other vessels of small dimensions, carrying heavy armaments, considerable protection, and good coal supplies."

	Torpedo boat.	Torpedo gunboat, *Sharpshooter* class.	*Medusa*, 3d class cruiser.	*Terpsichore*, 2d class cruiser.	*Edgar*, 1st class cruiser.	*Blenheim*, 1st class cruiser.	Atlantic passenger steamer.
Length (in feet)...	135	230	265	300	360	375	525
Breadth " ...	14	27	41	43	60	65	63
Draught (on trial)	5′ 1″	8′ 3″	16′ 6″	16′ 2″	23′ 9″	25′ 9″	21′ 3″
Disp. "	103	735	2,800	3,300	7,390	9,100	11,500
I. H. P., 10 knots..	110	450	700	800	1,000	1,500	2,000
I. H. P., 14 " ..	260	1,100	2,100	2,400	3,000	4,000	4,600
I. H. P., 18 " ..	870	2,500	6,400	6,000	7,500	9,000	10,000
I. H. P., 20 " ..	1,130	3,500	10,000	9,000	11,000	12,500	14,500

There is abundance of testimony on this point in our own ships. The *Yorktown* made a voyage of 10 days at 11.1 knots per hour on 32.8 tons of coal a day. Her displacement was about 1,775 and the bottom was somewhat foul. One ton of coal thus drove 54.1 tons weight through the water at 11.1 knots. The *San Francisco* made a voyage under similar conditions of 9 days on 45.4 tons coal at exactly the same speed. Her displacement was about 4,500 tons, which gives 99.1 tons weight driven 11.1 knots for one ton of coal. For this reason alone we can see how hopeless it is to try to make commerce destroyers of small size. Small displacement only supplies the objector and the newspapers with a club to wield against the new navy; it gives non-professional people ready means of making invidious comparisons between the performance of naval and merchant ships. The formula I. H. P. $= \frac{D^{\frac{2}{3}} \times S^3}{C}$ in common use among designers furnishes explanation of a great part of the above. It also shows clearly why a large ship has the advantage as to weight of machinery. One ton of machinery in the *San Fran-*

cisco will drive 5.14 tons weight at 19.6 knots, while one ton in the *Raleigh*, a vessel designed to make about 20 knots, will drive only 3.7 tons weight.

10th. A falling off occurs in some cases on account of the difference in boilers. Some ships have boilers that cannot be forced on a trial trip and yet which give excellent results at lower speeds. For this reason the results of measured-mile trials often lead to false conclusions in regard to sea steaming. I am inclined to think that the gunboat boilers in the *Yorktown*, *Bennington*, *Concord* and *Charleston* are not economical at low speeds although they have enormous steam surfaces and can be forced without the slightest foaming.

11th. Similar remarks apply to the screw. A screw which gives fine results on the measured-mile trial may be very poor in service. This was well shown in the *Medea* class in England where they actually used an inferior screw for the trials because it gave better results at lower speeds.

12th. A large surplus of power is required when keeping station in a fleet, and the inevitable accompaniment, loss of economy. A vessel, to make 10 knots steadily, should be capable of making 12 at times. The boiler power would thus have to be carried 75 per cent. over that absolutely necessary when steaming singly.

13th. Another great discrepancy arises from the employment of theoretical consumption of coal for low powers, based upon the trial trips of engines working near their maximum powers with the best coal; also in making no allowance for the great wastefulness of auxiliary machinery working at low speeds. It has become the fashion with many officers who do not have to look after the coal account to give the compound engine 2 pounds of coal per horse power per hour, and then, in the enthusiasm of the moment, to put down the triple-expansion engine at 1.50 pounds. As a matter of common experience, at low powers and ordinary cruising speed, the compound engine may be just as economical as a triple-expansion engine; in fact, there are cases where a simple engine might give the best results. While there is no question of the superiority and economy of

triple expansion worked up to the designed power on natural draft, it is by no means a settled question as to its superiority for cruising. In some cases, the low-pressure cylinder is designed to be disconnected in order to work the other two cylinders as compound, with very fair results, I believe. Some of the reasons for this will be more fully stated later on.

During the contractor's trials of three of the later ships, the coal was carefully weighed with the following results for the coal per hour per I.H.P.: *Newark*, 2.31 pounds; *Concord*, 2.76 pounds; *Bennington*, 2.60 pounds. This does not look much like 1.50 pounds. Commenting upon the results of these tests, the "Engineer" of January 22, 1892, says: "We see no reason to think that English engines would do any better, and we believe that consumption anywhere near 1.5 pounds per horse power per hour with the engines going at full speed exists only in the sanguine imaginations of a few. * * * We have at least the satisfaction of knowing that in all likelihood none of our neighbors are a bit better off than we are."

The *Charleston* never got down to 2 pounds of coal per horse power per hour during her entire cruise, and the *Concord* shows a very wide divergence from the tabulated calculation of 1.60 pounds. On the voyage mentioned before, the average amount of coal used per day was 22.4 tons, giving 832.9 total I.H.P., or 2.51 pounds per horse power per hour; taking out half a ton for galley and distilling, this becomes 2.45 pounds. It is difficult to convince officers that the waste of coal shown above does not lie in the fire room, inasmuch as merchantmen often make a better showing than the Navy, but a little consideration of the question in connection with the most economical speed will bring out the causes which lead to the low economy of our ships.

MOST ECONOMICAL SPEED.

All the waste arises from the fact that we design the ships for high speed and run them at low. Also, when the boilers are running natural draft and, therefore, economically, the engines are running at half power and wastefully; when the engines are running at high speed and with highest efficiency, the blowers

are running and a large percentage of the heat goes up to the smoke pipe. However, as the subject is closely involved in the following, I leave it here, to discuss more fully why the economical speed of a ship is not much lower than actual service reveals.

The table of the *Yorktown's* performance on the measured mile is added for this purpose. First, let me state that the most economical speed of a ship is that speed at which one ton of coal will drive her the greatest distance. This speed may vary in the same ship under different conditions of wind, sea, bottom and displacement. The above table for the *Charleston* demonstrates that, while the greatest distance is covered with one ton of coal at 51 revolutions, the actual speed varies from 6.8 with foul bottoms to 8.8 just out of dock.

Usually, the horse power is said to vary approximately with the cube of the speed, and a calculation for the *Yorktown's* endurance based upon that will give a much greater result than is realized in actual practice. The above law of horse powers is practically true when applied to the power utilized in driving the ship, but it is far from correct for the power developed by the coal, especially at low speeds where the useful work forms only a small percentage of the total work. The whole subject could be dismissed by stating an axiom among engineers that a machine becomes wasteful in proportion to its departure from the power for which it was designed, the condition under which all of our warships operate; but that does not make clear the actual causes of loss.

The distribution of power resulting from the combustion of coal in the furnaces is about as follows:

1st. The work applied to the screw, made up of thrust which propels the ship and losses from slip and friction of the blades in the water. The percentage of losses increases as the speed varies from the economical rate, shown plainly on the trial trip of the *Charleston*, by the percentage of slip increasing whether speed goes up or down. The above work is easily measured on an indicator card, by making due allowance for losses within the engines.

2d. An enormous amount of heat escapes up the smoke pipe,

greatly increased at high powers when the blowers drive the fire into the uptakes. Where the heating surface is small compared with the grate surface this waste forms a large percentage of the coal burnt.

3d. The waste from condensation in boilers and pipes, leaks and radiation. This is made evident when we measure the coal required to keep fires banked. It is readily determined, and in case of the *Yorktown* amounts to one ton a day for each boiler.

4th. The waste due to condensation in the cylinders and the drop in receivers. This increases enormously when engines of high power are put to small duty.

5th. The waste due to friction of the moving parts of an engine. Since this depends upon tightness of "fit" of the journals, valves and pistons, its value in horse power is proportional to the number of revolutions. The extent to which this affects the efficiency of an engine may be seen when we consider that the moving parts of the *Yorktown* are all made large enough for 3,662 I.H.P., and that the friction of these large parts must be overcome in driving the ship 4.4 knots. Its approximate value is easily determined from the jacking engine or the initial thrust of a speed curve.

6th. The power expanded on the auxiliary machinery before the main engines are moved at all. It is probably constant up to half speed, while the power of the main engines is low. A large amount of coal is absorbed in driving this machinery, much greater in proportion than the main engines require. The horse power for auxiliaries is obtained from indicator cards, and it is therefore fairly well known in every ship. In this case, also, the machinery is designed for high power and run at low.

In the above it will be seen that all elements except No. 1 count against the ship at high powers, and that all except Nos. 1 and 2 are enormous in comparison with the work exerted upon the screw at low powers.

I have made out a table for the *Yorktown* to show the most economical speed based upon calculations, making due allowances for all losses and auxiliaries.

The data given for 4.4, 10.62, 14.81 and 16.65 knots were

taken from the measured mile trials of 1889, excepting the coal expenditure. That for 10.62 knots is deduced from the expenditure for 96 hours under exactly similar conditions otherwise. The power for 6 and 8 knots and all other coal expenditures are obtained by calculations based on actual practical results. The coal per I.H.P. of the main engines alone is taken at 4 pounds per hour at 4.4 knots, 2.1 pounds at 10.62 and proportionally for other powers, the coal for auxiliaries ranging from 5 to 4 pounds, which experience proves to be rather low. (This expenditure is justified by results obtained from the *Charleston's* dynamo when running the auxiliary boiler on that alone; also from the air and circulating pumps when under banked fires.)

The table shows very clearly what is borne out by the log book, that there is a considerable range of speed (for the *Yorktown* from 7 to 9 knots), for which the endurance is nearly the same.

The curves have been constructed to show the results of the table at a glance. The upper curve represents the number of knots run per ton of coal, measuring the ordinates, the speed being laid out on the axis of X. From it we see that there are always two speeds at which the ship has the same endurance. The most economical speed is from 7.5 to 8.5 knots, and the ship will cover the same distance at 6 and 10.2 knots, but in much less time at the latter speed. This point should be of special interest to the navigator in making a port, and will enable him to decide whether it is better to slow down, or increase the speed and avoid wasting coal by lying off under banked fires all night. If the captain does not wish to enter a harbor just after nightfall and another boiler be necessary to increase the speed, it is always better to slow down, providing, of course, there is no emergency in the matter.

In the diagram, the lower curve shows the tons of coal burnt per hour for the different speeds. The point where the curve crosses the axis of Y gives the amount of coal burnt with fires banked and auxiliaries running. I cannot claim great accuracy for these curves, as there is no reliable information upon which to base the power and coal for the auxiliaries.

The voyage of the *Concord* from Montevideo to Barbados may be examined again with some profit in this connection. She made 4,219 miles on 419.75 tons of Cardiff coal, at the rate of 9.4 knots. The fore-and-aft sails carried probably about offset the barnacles on the bottom, and we get, as the result of the whole trip, 2.51 pounds coal per I.H.P. per hour. The result worked out for the *Yorktown* above is somewhat higher, but the auxiliary machinery of the latter is probably more expensive ; that being the only difference between the two ships, excepting the system of forced draft, which was not used on the voyage.

That there should have been many mistakes in the early calculations relating to the new Navy is perfectly natural, and I repeat that there is no intention to reflect upon the work of any Bureau. Our new ships are creditable additions to the Navy, but there is nothing gained by self-deception in regard to their performances. The only thing that surprises me is that officers should continue such deceptive methods in the light of experience. To the *Columbia* is given a radius of action of 26,240 miles and 109 days' endurance, when not one of our ships has realized two-thirds of the results obtained from similar calculations. The above is based on 10 knots and 18.3 tons of coal per day. The *Yorktown*, of 1,700 tons displacement, has never gotten below 20 tons for 10 knots, and here a ship of 7,400 tons displacement is to be propelled at that rate on 18.3 tons.

The following extract from the last report of the Engineer-in-Chief bears directly on this point :

" The results obtained from the accurate trials of the *Newark*, *Concord* and *Bennington* give the first trustworthy figures in regard to the coal required per horse power under forced draft full power that have ever been published. They show, conclusively, what this Bureau has steadily maintained, that the figures so often published in estimates of coal endurance are entirely too low, and that the radii of action predicated therefrom are entirely too high.

" While these results will undoubtedly be a source of regret to all who had mapped out fields of usefulness for our vessels, based on the more roseate view, there can be no doubt that the best interests of the Government have been subserved by finding out the truth. It is sure to be brought home to us some day, and it

is far better that we should know it now and act upon it by providing large coal capacity, than wait for the humiliation of a ship found to be helpless when we were most in need of her services.

"While on this subject of coal economy it may not be amiss to say a few words to clear up an impression, which seems to exist to some extent, that our vessels are decidedly uneconomical as compared with those of the merchant marine in ordinary cruising, and it is pointed out that our vessels need recoaling after short runs, while the merchant steamers make long continuous voyages. It is frankly admitted that our machinery is not as economical, but from the nature of things it cannot be. The circumstances of the two services are entirely dissimilar. The economical merchantman has small engines and boilers for a large hull, while our ships have powerful machinery for small hulls. This machinery must be built to develop the maximum power which will ever be required, while ordinary cruising is done at a fraction of this power, a circumstance which is inimical to economy.

"An interesting comparison of the conditions of the two cases is furnished by the *Baltimore* of our Navy and a merchant steamer called the *Iona*, which have about the same displacement—4,450 tons. The *Baltimore's* machinery develops 10,000 I.H.P.; that of the *Iona*, 700 I.H.P. The *Baltimore* has room for only 17,000 square feet of heating surface in her boilers, with a ratio to grate surface of less than 30, while for the *Iona* the figures are 3,160 and 75. The *Iona* works always at full power, and secures a speed of 8½ knots; for this same speed the *Baltimore* would require more power on account of the friction of the enormously larger engines. But the great economy is in the boilers. With the enormous amount of heating surface for the power developed the *Iona* can evaporate 10.5 pounds of water per pound of coal, while the *Baltimore* probably does not exceed 8 pounds. The *Baltimore's* boilers weigh 490 tons for 10,000 I.H.P., and the *Iona's* 122 tons for 700 I.H.P. Were the *Baltimore's* boilers built for economy instead of power on the same ratio as the *Iona's*, they would weigh 1,743 tons, or nearly twice as much as the entire machinery of the *Baltimore* does. The comparison was made between these two ships purposely, because they are supposed to be of best English design.

"In other words, economical machinery means heavy machinery taking up much room; but the power required in swift war vessels of moderate size is so great that to make the machinery both powerful and economical the whole ship would have to be given up to it. As the great requisite in our ships

is powerful, but light machinery, economy must, of necessity, be sacrificed."

It may be of interest, while on this subject of speed, to recall the fact that the Engineer-in-Chief has suggested the use of the screw in determining the speed of a ship on the contract trial. It should be standardized by running over the measured mile in opposite directions a sufficient number of times under the conditions required by the contract, to determine the number of revolutions to the knot for several different speeds. This result could be used in subsequent determinations of speed, thus eliminating all questions of current, course and sea. The accuracy of the screw is probably much greater than that of the log, as its surfaces are enormously larger, and the blades are not likely to bend or leap out of the water. With a measured-mile course of forty miles there is always inaccuracy, both in the exact distance and the allowance for currents. As an illustration of the patent log, the case of the *Boston* and *Atlanta* steaming together from Rio de Janeiro to St. Lucia presents some interesting points. The *Atlanta* made 3,076 miles by the log, and the *Boston* 3,174, just 98 miles farther.

In conclusion, I will say that we have no cause to be dissatisfied with our new ships. A few cases have been selected from the performance of the English ships to show that we have done at least as well, if not better.

The *Melpomene* is put down for 8,000 knots steaming radius. When bound from St. Vincent to Montevideo she had to put into Bahia for coal, showing the steaming radius to be nearer 3,000 knots. She was unable to maintain 15 knots' speed for 200 knots, although designed to make over 19.

The *Medea* exceeded 19 knots on her trial trip. She did not reach 16 on a full power run from the English coast to Gibraltar.

In the manœuvers of 1889 the *Mercury* was the only ship in an entire squadron capable of making 16 knots for 24 hours.

The *Mersey* reached only 14.5 knots from Bantry Bay to Lough Swilly when sent on dispatch duty, and yet she was obliged to slow down several hours to sweep tubes and get things to rights.

Table V.

U. S. S. *CONCORD.*

DISPLACEMENT = 1,703 TONS. MAXIMUM SPEED = 16.8. BUNKERS = 401 TONS ANTHRACITE.

STEAMING RADIUS AND ENDURANCE FROM THE STEAM LOG.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
No.	Knots speed per hour.	Revolutions per minute.	Revolutions per knot per minute.	Revolutions per knot.	Knots per day.	Tons coal per day.	Knots per ton of coal.	Days steaming.	Knots steaming radius.	Coal. Kind.	Coal. Quality.	General direction of wind.	Kind of sea.	Months out of dock.	Condition of bottom.	No. days in voyage taken.	No. boilers used.
a 1	8.6	79 9	9.29	557	206.6	19.2	10.76	20 9	4.315	B	Good	Beam—light	Smooth	3	Rough	12	3
b 2	9 4	86.1	9 16	550	225.0	22.4	10 05	17 9	4.030	B	Good	Var.—light	Smooth	4	Foul	19	3
c 3	10 1	88.6	8 77	526	242.0	20.3	11 92	19.7	4.780	B	Good	Beam—light	Smooth	3	Rough	6	3
d 4	11.7	95.4	8.15	489	279.4	27.1	10.31	14.8	4,134	B	Good	Beam 1–6	Smooth	1	Clean	5	3

a Fore-and-aft sails 7 days. *b* Making best use of sails possible. *c* Fore-and-aft sails 5 days. *d* Fore-and-aft sails all the voyage.

Steaming radius with only absolutely necessary expenditures for auxiliaries.—Knots, speed per hour, 8.2; knots, steaming radius, 5,130.

Expenditures for galley, dynamo, distilling, &c., per day = 1.5 tons.

Economical revolutions = 80 per miuute. Speed = 8.2 knots with clean bottom.

The longest run made by this ship was in February, 1892, from Montevideo to Barbados, a distance of 4,219 knots by the log, which she made in 18 days, 18 hours, at the rate of 9.4 knots under three-fourths boiler power, natural draft. The total amount of coal consumed was 420 tons, of good bituminous, or 22.4 tons per day. Weather generally favorable. Sails set 12 days, with wind on beam; force, 3; bottom, rough.

Table VI.

U. S. S. *BENNINGTON.*

DISPLACEMENT = 1703 TONS. MAXIMUM SPEED = 17.5 KNOTS. BUNKERS = 403 TONS ANTHRACITE.

STEAMING RADIUS AND ENDURANCE, FROM THE STEAM LOG.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
No.	Knots speed per hour.	Revolutions per minute.	Revolutions per knot per minute.	Revolutions per knot.	Knots per day.	Tons coal per day.	Knots per ton of coal.	Days steaming.	Knots steaming radius.	Coal. Kind.	Coal. Quality.	General direction of wind.	Kind of sea.	Months out of dock.	Condition of bottom.	No. days in voyage taken.	No. boilers used.
a 1	8.8	74.9	8.51	511	210.1	21.5	9.78	18.7	3941	B	Bad	Var.—light	Smooth	1½	Clean	5	3
b 2	9.4	79.2	8.43	505	225.8	18.7	12.07	21.5	4864	A	Good	Var.—Var.	Mod.	1	Clean	11	3
c 3	9.9	90.9	9.09	545	233.3	23.1	10.10	17.5	4070	B	Good	Var.—Var.	Mod.	2	Clean	6	3

a Fore-and-aft sails, 3½ days. *b* Fore-and-aft sails, 2½ days. Strong wind on beam, 2½ days. *c* Fore-and-aft sails, 4½ days.

Steaming radius, with only absolutely necessary expenditures for auxiliaries.—Knots speed per hour, 8.2; knots steaming radius, 5130.

Expenditure for galley, dynamo, distilling, &c., per day = 1.5 tons.

Economical revolutions = 80 per min. Speed = 8.2 knots, with clean bottom.

Table VII.

U. S. S. *BALTIMORE.*

DISPLACEMENT = 4,400 TONS. MAXIMUM SPEED = 19 58. BUNKERS = 1,144 TONS ANTHRACITE.

STEAMING RADIUS AND ENDURANCE FROM THE STEAM LOG.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
										Coal.							
No.	Knots speed per hour.	Revolutions per minute.	Revolutions per knot per minute.	Revolutions per knot.	Knots per day.	Tons coal per day.	Knots per ton of coal.	Days steaming.	Knots steaming radius.	Kind.	Quality.	General direction of wind.	Kind of sea.	Months out of dock.	Condition of bottom.	No. days in voyage taken.	No. boilers used.
a 1	8.7	52.3	6.01	366	209.9	43.8	4.80	26.11	5491	B	Good	Var.—light	Smooth	7	Rough	10	2
b 2	9.65	57.1	5.92	355	231.5	42.0	5.51	27.24	6306	B	Good	Light	Smooth	10	Rough	17	2
3	11.7	67.7	5.79	347	281.1	62.5	4.50	18.30	5144	B	Good	Beam—light	Smooth	10	Rough	2	4
4	10.0	55.2	5.52	331	239.5	45.0	5.32	25.42	6089	B	Good	Beam—var.	Smooth	4	Rough	20	2
5	11.3	60.5	5.35	321	271.5	46.7	5.81	24.49	6651	B	Best	Beam—var.	Smooth	½	Clean	17	2
c 6	13.0	68.9	5.30	318	311.9	74.2	4.20	15.42	4809	B	Best	Beam—var.	Mod.	4	Rough	6	4

a 1 day, strong head wind. *b* 1 day, strong head wind. *c* 2 days, heavy gale and sea on beam.

Steaming radius with only absolutely necessary expenditures for auxiliaries.—Knots, speed per hour, 11.3; knots per day, 271.5; tons coal per day, 44; knots per ton of coal, 6.17; knots steaming radius, 7,058.

Expenditures for galley, dynamo, distilling, &c., per day = 3.9 tons.

Economical revolutions = 57 per miuute. Speed = 10.6 knots with clean bottom.

The longest run made by the ship was in August, 1890, from New York to Stockholm, a distance of 4,677 knots by the log, which she made in 19 days 12 hours, at the rate of 10 knots, under half boiler power, natural draft. The total amount of coal consumed was 878 tons of good bituminous, or 45 tons per day. Weather generally favorable and the bottom clean.

Table VIII.

U. S. S. *PHILADELPHIA.*

DISPLACEMENT = 4,325 TONS. MAXIMUM SPEED = 19.68 KNOTS. BUNKERS = 1,032 TONS ANTHRACITE.

STEAMING RADIUS AND ENDURANCE, FROM THE STEAM LOG.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
										Coal.							
No.	Knots speed per hour.	Revolutions per minute.	Revolutions per knot per minute.	Revolutions per knot.	Knots per day.	Tons coal per day.	Knots per ton of coal.	Days steaming.	Knots steaming radius.	Kind.	Quality.	General direction of wind.	Kind of sea.	Months out of dock.	Condition of bottom.	No. days in voyage taken.	No. boilers used.
1	9.3	57.8	6.22	373	223.5	40.5	5.52	25.48	5696	B	Good	Var.—light	Smooth	5	Foul	19	2
2	10.0	59.7	5.97	358	234.0	42.8	5.47	24.12	5644	B	Good	Var.—light	Smooth	4	Foul	19	2
3	12.1	74.3	6.14	368	290.9	74.0	3.93	13.95	4057	B	Good	Beam—strong	Mod.	5	Foul	5	3½
4	11.6	68.0	5.86	352	278.	44.1	6.30	23.40	6506	A	Good	Beam—strong	Mod.	1	Clean	5	3
5	12.0	72.6	6.05	363	286.8	63.7	4.50	16.20	4646	B	Good	Beam—strong	Mod.	1	Clean	5	3

Steaming radius, with only absolutely necessary expenditures for auxilaries.—Knots speed per hour, 11.6; knots per day, 278; tons coal per day, 41.3; knots per ton of coal, 6.73; knots steaming radius, 6,945.

Expenditures for galley, dynamo, distilling, &c., per day = 3.9 tons.

Probable economical revolutions = 60 per min. Speed = 10.2 knots, with clean bottom.

The longest run made by this ship was in January and February, 1892, from Bridgetown to Montevideo, a distance of 4,481 knots by the log, which she made in 19 days 4 hours, at the rate of 10 knots, under half boiler power, natural draft. The total amount of coal used was 819 tons of good bituminous, or 42.8 per day. Weather generally favorable. Bottom somewhat foul.

Table IX.

U. S. S. *SAN FRANCISCO.*

DISPLACEMENT = 4,088 TONS. MAXIMUM SPEED = 19.53 KNOTS. BUNKERS = 628 TONS ANTHRACITE.

STEAMING RADIUS AND ENDURANCE FROM THE STEAM LOG.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
										Coal.							
No.	Knots speed per hour.	Revolutions per minute.	Revolutions per knot per minute.	Revolutions per knot.	Knots per day.	Tons coal per day.	Knots per ton of coal.	Days steaming.	Knots steaming radius.	Kind.	Quality.	General direction of wind.	Kind of sea.	Months out of dock.	Condition of bottom.	No. days in voyage taken.	No. boilers used.
1	9.14	61.8	6.76	405	219.4	40.3	5.44	15.58	3419	B	Best	Var.—light	Smooth	7½	Rough	22	2
2	9.6	58.8	6.12	367	234.8	34.8	6.75	18.05	4237	A&B	Good	Var.—light	Smooth	1	Clean	7.5	2
3	10.5	60.0	5.71	343	252.3	37.2	6.78	16.80	4259	A	Good	Var.—light	Smooth	1	Clean	7.5	2
4	11.1	65.2	5.87	352	266.5	45.4	5.87	13.83	3686	B	Good	Var.—light	Smooth	3	Clean	8.6	2

Steaming radius with only absolutely necessary expenditures for auxiliary purposes.—Knots speed per hour, 10.5; knots per day, 252.3; tons coal per day, 35.2; knots steaming radius, 4,521.

Expenditures for galley, dynamo, distilling, &c., per day = 3.2 tons.

Economical revolutions = 60 per minute. Speed = 10.5 knots, with clean bottom.

The longest run made by this ship was in February, 1892, from San Diego to Honolulu, a distance of 2,292 knots by the log, which she made in 8 days 15 hours, at the rate of 11.1 knots, under half-boiler power, natural draft. The total amount of coal consumed was 390 tons of anthracite, or 45.4 tons per day. The weather was favorable and the bottom clean.

Table X.

U. S. S. *NEWARK.*

DISPLACEMENT = 4,098 TONS. MAXIMUM SPEED = 19 (?) KNOTS. BUNKERS = 810 TONS ANTHRACITE.

STEAMING RADIUS AND ENDURANCE FROM THE STEAM LOG.

1 No.	2 Knots speed per hour.	3 Revolutions per minute.	4 Revolutions per knot per minute.	5 Revolutions per knot.	6 Knots per day.	7 Tons coal per day.	8 Knots per ton of coal.	9 Days steaming.	10 Knots steaming radius.	11 Coal. Kind.	12 Coal. Quality.	13 General direction of wind.	14 Kind of sea.	15 Months out of dock.	16 Condition of bottom.	17 No. days in voyage taken.	18 No. boilers used.
a 1	10.0	65.5	6.55	393	238.7	41.4	5.77	19.57	4671	B	Good	Var.—light	Smooth	4½	Rough	6	2

a Sails to top-gallants, fore-and-aft sails, 2 days.

Expenditures for galley, dynamo, distilling, &c., per day = 3.9 tons.

Probable economical revolutions = 60 per miuute.

This ship had made no long run up to June 30th, 1892, and the above steaming radius at 10 knots is based on the voyage from La Guayra to Key West, under favorable conditions, but with bottom somewhat foul.

Table XI.

U. S. S. *CHICAGO.*

DISPLACEMENT = 4 530 TONS. MAXIMUM SPEED = 15.33 KNOTS. BUNKERS = 832 TONS ANTHRACITE.

STEAMING RADIUS AND ENDURANCE, FROM THE STEAM LOG.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
										Coal.							
No.	Knots speed per hour.	Revolutions per minute.	Revolutions per knot per minute.	Revolutions per knot.	Knots per day.	Tons coal per day.	Knots per ton of coal	Days steaming.	Knots steaming radius.	Kind.	Quality.	General direction of wind.	Kind of sea.	Months out of dock.	Condition of bottom.	No. days in voyage taken.	No. boilers used.
a 1	9.4	41.5	4.41	265	226.6	49.3	4.60	16.9	3827	A	Poor	Bow—light.	Smooth	1	Clean	4	3
b 2	9.5	41.4	4.36	262	228.3	40.7	5.61	20.4	4668	B	Best	Beam—1—5	Smooth	1	Clean	18	4
c 3	10.6	45.0	4.24	254	253.8	46.0	5.52	18.1	4593	B	Best	Beam—1—5	Smooth	3	Clean	6	3

a No sails. *b* Sails, 12 days. *c* Sails, 4 days.

Steaming radius, with only absolutely necessary expenditures for auxilaries.—Knots speed per hour, 9.5; revolutions per minute, 41.4; knots per day, 228; tons coal per day, 37.7; knots per ton of coal, 6.06; days steaming, 22.1; knots steaming radius, 5,042, with sails.

Expenditures for galley, dynamo, distilling, &c., per day = 4 tons.

Economical revolutions = 41 per min. Speed = 9.5 knots, with clean bottom.

The longest run made by this ship was in December, 1891, from Barbados to Montevideo, a distance of 4,179 knots by the log, which she made in 18 days 7 hours, at the rate of 9.5 knots, under two-thirds boiler power, natural draft. The total amount of coal used was 744 tons of good bituminous, or 40.7 per day. Weather favorable and sails set 12 days. Bottom fairly clean.

Table XII.

U. S. S. *BOSTON.*

DISPLACEMENT = 3,030 TONS. MAXIMUM SPEED = 15.59 KNOTS. BUNKERS = 536 TONS ANTHRACITE.

STEAMING RADIUS AND ENDURANCE FROM THE STEAM LOG.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
No.	Knots speed per hour.	Revolutions per minute.	Revolutions per knot per minute.	Revolutions per knot.	Knots per day.	Tons coal per day.	Knots per ton of coal.	Days steaming.	Knots steaming radius.	Coal. Kind.	Coal. Quality.	General direction of wind.	Kind of sea.	Months out of dock.	Condition of bottom.	No. days in voyage taken.	No. boilers used.
a 1	8.3	39.8	4.79	287	197.1	36.8	5.37	14.6	2880	B	Good	Var.—light	Smooth	...	Rough	12	5

a No sails.

Steaming radius with only absolutely necessary expenditures for auxiliary purposes.—Knots speed per hour, 8.4; revolutions per minute, 39.0; knots per day, 201; tons coal per day, 24; knots per ton of coal, 8.38; days steaming, 21.4; knots steaming radius, 4,492 with sails.

Expenditures for galley, dynamo, distilling, &c., per day = 2.8 tons.

Economical revolutions = 39 per minute. Speed = 8.4 knots, with clean bottom.

The longest run made by this ship was in July, 1890, from Rio Janeiro to St. Lucia, a distance of 3,174 knots by the log, which she made in 14 days 20 hours, at the rate of 8.94 knots under one half boiler power, natural draft. The total amount of coal used was 386 tons of good bituminous or 26.1 tons per day. Weather favorable and sails set 12 days. The bottom was fairly clean.

Table XIII.

U. S. S. *ATLANTA.*

DISPLACEMENT = 3,030 TONS. MAXIMUM SPEED = 15.5 KNOTS. BUNKERS = 490 TONS ANTHRACITE.

STEAMING RADIUS AND ENDURANCE, FROM THE STEAM LOG.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
No.	Knots speed per hour.	Revolutions per minute.	Revolutions per knot per minute.	Revolutions per knot.	Knots per day.	Tons coal per day.	Knots per ton of coal	Days steaming.	Knots steaming radius.	Coal. Kind.	Coal. Quality.	General direction of wind.	Kind of sea.	Months out of dock.	Condition of bottom.	No. days in voyage taken.	No. boilers used.
a 1	8.75	41.6	4.76	285	210.1	27.1	7.75	18.1	3799	B	Good	Quar.—3—5	Smooth	3	Clean	9	4
b 2	9.48	45.1	4.76	285	232.3	36.6	6.35	13.4	3110	A&B	Mixed	Beam—1—5	Smooth	1	Clean	5	4—6
c 3	10.15	47.7	4.70	282	243.5	32.8	7.42	14.9	3638	B	Good	Beam—3—5	Smooth	3	Clean	4	4

a Topgallant, fore-and-aft sails. *b* No sails. *c* Topgallant, fore-and-aft sails.

Steaming radius, with only absolutely necessary expenditures for auxilaries.—Knots speed per hour, 8.75; revolutions per minute, 41.6; knots per day, 210.1; tons coal per day, 25.1; knots per ton of coal, 8 37; days steaming, 19.5; knots steaming radius, 4,101, with sails.

Expendutures for galley, dynamo, distilling, &c., per day = 2.8 tons.

Economical revolutions = 40 per min. Speed = 8.4 knots, with clean bottom.

The longest run made by this ship was in July, 1890, from Rio Janeiro to St. Lucia, a distance of 3,076 knots by the log, which she made in 14 days 20 hours, at the rate of 8.64 knots, under one-half and five-eighths boiler power, natural draft. The total coal consumption was 412 tons of good bituminous, 27.8 tons per day. Weather favorable and sails set 12 days. Bottom fairly clean.

Table XIV.

U. S. S. *PETREL.*

DISPLACEMENT = 855 TONS. MAXIMUM SPEED = 11.79 KNOTS. BUNKERS = 200 TONS ANTHRACITE.

STEAMING RADIUS AND ENDURANCE, FROM THE STEAM LOG.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
No.	Knots speed per hour.	Revolutions per minute.	Revolutions per knot per minute.	Revolutions per knot.	Knots per day.	Tons coal per day.	Knots per ton of coal	Days steaming.	Knots steaming radius.	Coal. Kind.	Coal. Quality.	General direction of wind.	Kind of sea.	Months out of dock.	Condition of bottom.	No. days in voyage taken.	No. boilers used.
a 1	6.9	68.2	9.88	593	165.1	8.0	20.64	25.	4128	B	Good	Var.—strong	Mod.	2	Clean	34	1
b 2	10.5	92.3	8.79	527	251.0	11.65	21.55	17.2	4310	B	Good	Aft—2—5	Mod.	3	Clean	5	2

a Sails 5 days; gale 7 days; hove-to 2 days. *b* Sails few hours.

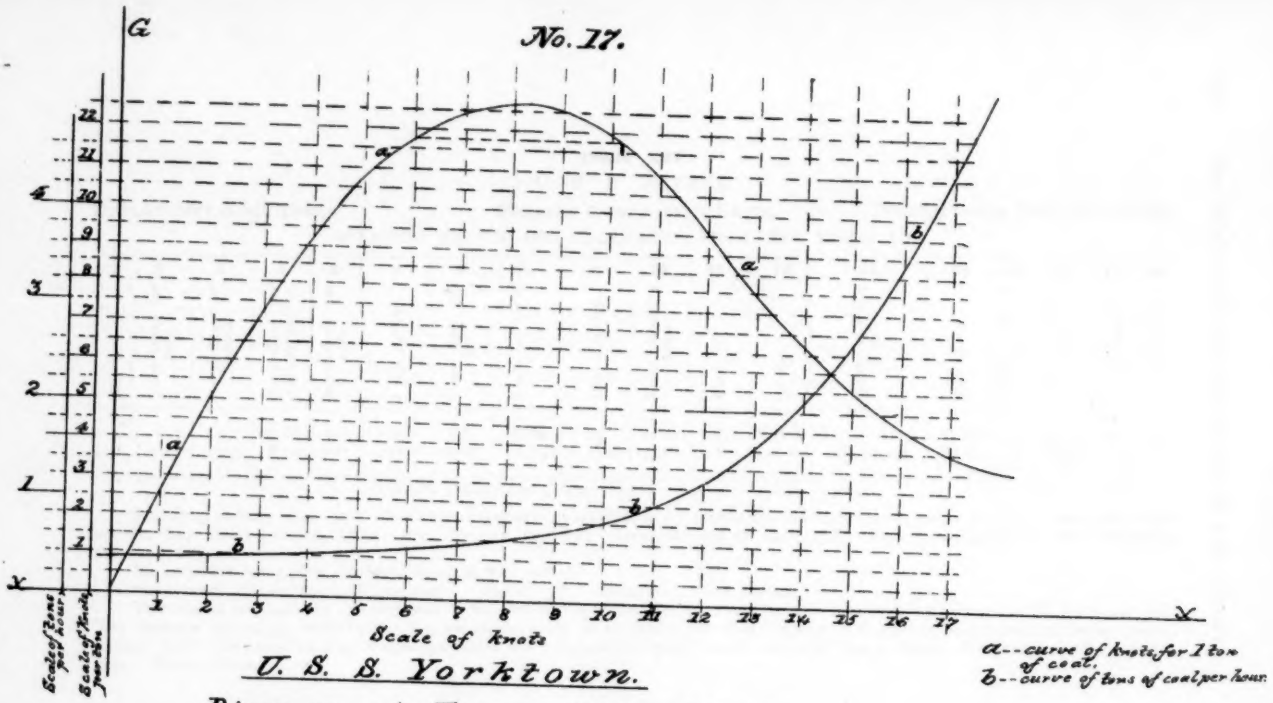
Steaming radius, with only absolutely necessary expenditures for auxiliaries.—Knots speed per hour, 7.6; revolutions per minute, 68; knots per day, 182.4; tons coal per day, 8.0; knots per ton of coal, 22 8; days steaming, 25.0; knots steaming radius, 4,560.

Expenditures for galley, dynamo, distilling, &c., per day = 1 ton.

Economical revolutions = 68 per min. Speed = 7.6 knots, with clean bottom.

The longest run made by this ship was in November, 1891, from Sandy Hook to Gibralter, stopping 36 hours at St. Lucia for coal. The distance was 5,333 miles by the log, which she made in 32 days 7 hours, at the rate of 6.9 knots under one-half boiler power, natural draft. The total amount of coal consumed was 258.7 tons of good bituminous, or 8 tons per day. Weather generally unfavorable. Bottom clean.

No. 17.



U. S. S. Yorktown.

Diagram of Economic Qualities.

Table XVI.

U. S. S. *YORKTOWN.*

CALCULATION OF ECONOMICAL SPEED AND POUNDS COAL PER I.H.P.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
No.	Speed, knots.	Revolutions per minute.	Per cent. slip.	I.H.P., main engine.	I.H.P., auxiliaries.	I.H.P., total.	I.H.P., friction.	I.H.P., screw.	Per cent. power on screw.	Coal per I.H P. pr. hr. main engs.	Coal per hour, main engines.	Coal per I.H.P., auxiliaries.	Coal per hour, auxiliaries.	Coal per hour, losses.	Coal per hour, outside.	Coal per hour, total.	Knots per ton.	Coal per I.H.P., per hour, lbs.	Coal, tons per hr.
1	4.4	39.5	9.64	75.23	80.50	155.13	33.80	41.43	26.6	4.0	300.9	5	402.5	186.6	100	990.	9.96	6.35	.44
2	6.0	54.1	10.10	151.13	80.50	231.63	46.10	105.03	45.4	3.25	493.2	4.5	362.3	186.6	100	1142.1	11.77	4.93	.51
3	8.0	72.3	10.70	310.40	80.50	390.90	61.40	249.00	63.7	2.60	807.0	4	322.0	186.6	100	1415.6	12.65	3.62	.63
4	10.62	96.0	12.00	661.11	80.50	741.61	81.56	579.55	78.1	2.10	1388.3	4	322.0	279.9	100	2090.2	11.38	2.82	.90
5	14.81	140.6	14.58	2216.93	120.00	2336.93	113.70	2103.23	90.0	1.90	4212.2	4	480.0	373.2	100	5165.4	6.42	2.33	2.30
6	16.65	160.8	16.07	3415.73	246.20	3661.93	127.80	3287.93	89.8	2.20	7514.6	4	984.8	373.2	100	8972.6	4.16	2.63	4.04

NOTE.—The lowest speed could be obtained with one boiler or a smaller expenditure of coal, but two would always be used at sea; hence the large allowance for losses.

The coal used at highest speed is probably somewhat greater in practice by the increased amount lost up the smoke-pipe.

In 15 the losses are due to condensation, radiation, &c., and the amount is that required to keep boilers under banked fires without doing any work.

In 16 the outside work includes the dynamos, distilling, galley, ventilating fans, &c.

The most economical speed here appears to be 8.5 knots per hour.

COAL EXPENDITURE FOR AUXILIARIES.

The coal expenditure at low speeds is so important a matter that I have worked out a table of expenditures for the *Charleston* in order to show where every pound of coal goes and wherein lie the opportunities of greater economy. The importance of the subject warrants some repetition, and I have not hesitated to restate some things in a different form. The data at hand are very meager. The accurate determination of quantities of steam used for auxiliary engines, and in general for all auxiliary purposes on board ship at different powers of the main engines, would supply information from which the economical speed of the ship might be calculated even before a run at sea. It is worth a thorough trial on one of the new ships now building. So far, knowledge of the horse powers of auxiliaries is in a chaotic state. The official trials, of course, determine their maximum values, but the intermediate powers while cruising are estimated without much approximation to accuracy. The coal expended on auxiliaries will undoubtedly be found far in excess of the most generous allowances. I have calculated the steam used per horse power per hour for every auxiliary engine from the terminal pressures, and have added 15 per cent. for initial condensation and entrained water, and have found 40 pounds the minimum expenditure of water for any one of them. It is only fair to say that the auxiliaries of the *Charleston* all have simple engines which are probably expensive, and other ships using compound engines might do better. In the explanations of the columns of the table, I give the dimensions of the engines and all the data from which the horse powers are obtained. Much of it will be found in a very accurate report of the trial trip supplied by Passed Assistant Engineer Warburton for this JOURNAL. None of my remarks must be construed as a criticism of the builders and inspectors of the ship; for the design bought in England was faithfully carried out with very few modifications, and the work well done.

Column 2—Main Engines.—The main engines are compound, with cylinders of 44 inches and 85 inches diameter, by 36 inches stroke. They are nearly horizontal, with a slight pitch towards

the shafts. The piston packing rings are 4½ inches broad, with a brass shoe under the lowest part, and the pistons go down after very little running, so that it is difficult to keep them steam tight. The cylinders have two piston valves, each fitted with narrow steel rings. These rings wear rapidly and unevenly, and allow a great waste of steam from leakage directly into the condenser.

The horse powers of the main engines are obtained from cards taken when the ship was at or near normal draught, and the speed was carefully observed. For 18.21 knots, we have the trial trip data; for 15.7 knots, the average of 24 hours was taken, and cards with the same revolutions, steam pressure, &c.; for 11 and 8.77 knots, the average of 4 consecutive days was taken with cards every day; for 4 and 6.7 knots, special cards were taken while the ship was steaming slowly, and the speed carefully obtained from the log. I give the results in the accompanying table. A speed curve was plotted from them which proved

Table XVIII.

SPEED AND I. H. P. FOR U. S. S. *CHARLESTON.*

Revolutions.	Knots.—Speed.	Main Engines. I. H. P.	Coal per day.
114.65	18.21	6,316	
97.50	15.70	4,099	
64.90	11.00	1,288	
52.10	8.77	653	32.25
39.50	6.70	342	
23.50	4.00	107	

to be remarkably fair, and the powers for all other speeds measured on the curve. No auxiliaries are included. The coal for 8.77 knots is the only reliable data on coal expenditure obtainable for any of the above speeds.

(I may say, in this connection, that a real service would be rendered by any commanding officer and chief engineer who would systematically arrange for runs of at least a week each, at speeds increasing by a knot per hour, from the lowest practicable speed up to that due to full power with natural draft, during which runs the coal would be accurately measured.

We should then have data on whose accuracy we could depend.)

These horse powers include all the powers for which coal was expended on the main engines—the friction of moving parts, the friction of screw, the loss from slip, the churning of water and the useful thrust.

Column 3.—The coal expended on the main engines alone is probably least per horse power when the engines are driven at their maximum power without being forced. As the speed decreases, the percentage of loss from condensation must increase. Even supposing that the condensation on the walls of the cylinder is constant for a given time and is independent of the revolutions, the percentage of loss would increase as the powers grew smaller and, hence, the coal expenditures would increase correspondingly. The drop in the receiver also adds to the loss at low speeds. On the other hand, when the engines are forced, some water is lifted and the steam is probably not used economically by reason of the small amount of expansion. I have not introduced the loss from leakage here, as its value cannot be estimated without knowing the exact condition of the packing rings. It is given in another place. The expenditure of 1.90 pounds of coal per hour at 16 knots increasing to 4 pounds at 4 knots and to 2.50 pounds at 18 knots, is warranted both by the cards and data of several other ships that have been examined.

Column 4—Air Pumps.—The *Charleston's* air pumps are double and vertical, with a shaft and heavy fly wheel placed above the steam cylinders. The cranks are nearly opposite. The steam cylinders have a diameter of 10 inches; the pump cylinders 24 inches, with a stroke of 16 inches for all pistons. The engines are simple, with no expansion (in order to start the pump easily), and the exhaust somewhat choked. In the following table the data of the first six lines are obtained from cards taken on the trial trip, the last from cards taken during the cruise.

In the last case cards were also taken from the pump cylinders, which gave a useful horse power of 10, leaving 20 as the horse power required to overcome the friction of the mechan-

Table XIX.

REVOLUTIONS AND I.H.P. OF AIR PUMPS.

Revolutions of air pump.	I.H.P. air pumps.	I.H.P. main engines.	Weight of steam per hour per I.H.P. of air pumps from terminal pressure.
142.	51.37	6,374	69 lbs.
140.5	48.83	5,995	
131.5	43.28	6,178	
130.0	41.11	6,106	
128.	49.36	6,331	
123.5	38.18	6,278	
71.	30.00	2,147	52.5 lbs.

NOTE.—Data include pumps for both engines.

ism. This should be very little below the power for the pumps without the main engines running, as is shown to be the case by another method of calculation.

The work done by the steam cylinders of the air pumps is made up of three component parts:

(1.) The useful work, dependent upon the amount of steam used by the main engine and auxiliaries. This must be nearly proportional to the power of the main engine, and it is taken to be so in the table.

(2.) The removal of air which leaks into the condenser through stuffing boxes, which goes into the boiler with the salt feed and which the feed pump carries in when it loses suction. The amount of air is probably greater at low speeds with a low mean pressure.

(3.) The friction of the engine itself. This probably varies as the revolutions, and the power to overcome it should be greatest at the highest revolutions of the pump. Inasmuch as the work of removing air, spoken of above, increases at a low speed, the decrease of friction work is probably offset. I have, therefore, taken the work due to the last two factors as constant at all speeds, and the calculations resulting will be sufficiently accurate for all purposes. At full power, on the trial trip, the I.H.P. of the main engine was 6,316, that of the air pumps 42.85.

At cruising speed, the same powers were 2,147 and 30 respectively.

Let H be the constant power of the pump with engines ready to start, and k a coefficient for the increase dependent upon the main engines; then,

$$H + 6{,}316\ k = 42.85$$
$$H + 2{,}147\ k = 30.00,$$

and we have $H = 23.34$, $k = .0033$, from which the horse powers set down in column 4 of the table are supplied. The quantity of water used per I.H.P. per hour, determined from the terminal pressures of twelve cards, is 60 pounds, to which 15 per cent. has been added. The weight of water seems to be less for the lower power of the pump, but there is probably increased condensation whose amount is unknown.

Column 5—Circulating Pump.—The circulating pump, which is centrifugal, is driven by one cylinder of 9-inch diameter by 8-inch stroke. The fan or runner has a diameter of 32 inches with a breadth of opening of 2 inches and an inlet of 14 inches in diameter. The blades are curved back, and water is taken in on both sides of the runner which projects half way into the discharge pipe. The following table was made out from cards taken after the trial trip with the pump run at different speeds. The horse powers have received slight corrections to make the curve accompanying the table a fair one.

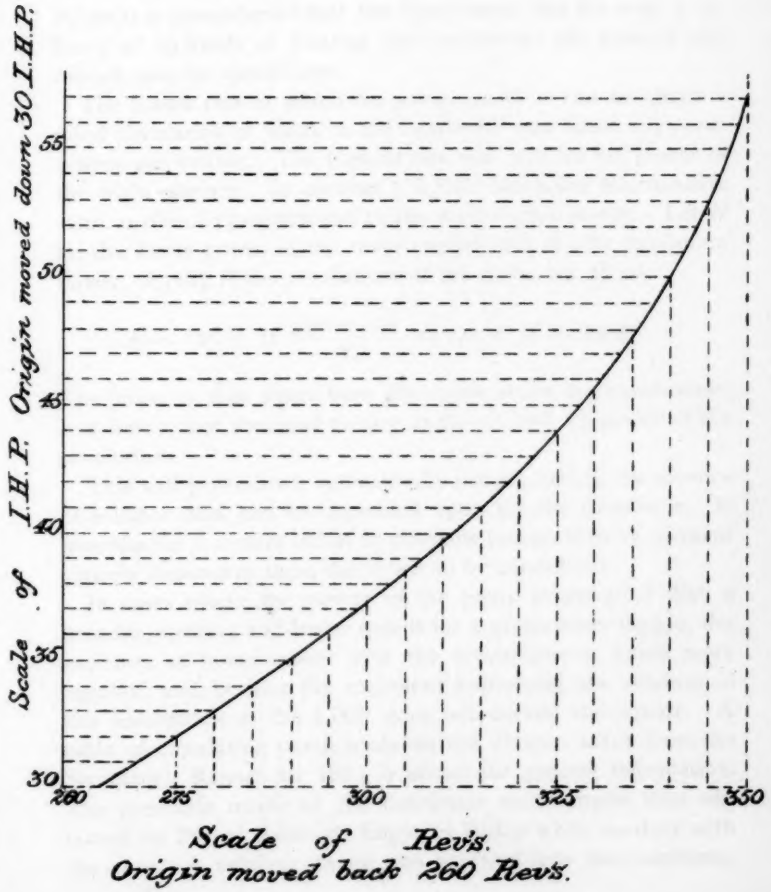
Table XX.

DATA FOR ONE CIRCULATING PUMP.

Revolutions per minute.	I.H.P. of one pump.	Exponential ratio of power to revolutions.	Pounds of water per I.H.P.
275	32.0		32.7
		1.76	
300	37.3		32.3
		1.94	
315	41.0		31.7
		2.26	
325	44.0		31.8
		2.83	
340	50.0		31.2
		3.80	
345	52.9		33.5
		5.63	
350	57.3		33.3

Average weight of water per I.H.P. per hour = 37.21 pounds.

Power Curve of Circulating Pump of U.S.S. Charleston.



I cannot satisfactorily account for this sudden and enormous increase of power at the higher speeds. Where a centrifugal pump is designed for a given speed, there is probably great increase of friction and decrease of efficiency at speeds exceeding it very much; and this is probably the trouble here. The opening through the fan is too narrow, and the projection of the fan into the discharge pipe makes the water difficult to get rid of. When it is remembered that the water leaves the fan with a velocity of 29 knots an hour at 350 revolutions, the force of this remark may be appreciated.

The lowest rate at which the pump could be run to obtain a good circulation of water in the condenser was about 125 revolutions per minute. The highest rate was 350 for full power of the main engines. In column 5, I have taken the intermediate rates as directly proportional to the main engine power. Let H be the horse power of the main engine, and H_m the maximum power, (6,316), R the revolutions of the pump fan, then—

$$R = 125 + H\,\frac{350 - 125}{H_m} = 125 + H \times .0356.$$

The power is then taken from the curve above 275 revolutions, and, below that, assumed to vary as the 1.6 and 1.5 power of the revolutions.

This will give results substantially correct, and, in the absence of reliable data, can be depended upon for the *Charleston*. It provides for a certain initial or constant power, with an increase entirely dependent upon the steam to be condensed.

In cases where the runner of the pump is designed with a broader opening and lower speeds for a given water supply, the variation of horse power with the revolutions is much more regular; and, besides, the exponent expressing the relation of the revolutions to the I.H.P. does not exceed the square. A table of circulating pump trials on the *Trenton*, taken from the Secretary's Report for 1883, is added for general information. The pressures inside of the condenser water chests were obtained by Passed Assistant Engineer Bailey while on duty with the ship. A receiver gauge was screwed into the condenser

head for that purpose. The velocity of the water leaving the runner was only 20 knots an hour at the highest speed.

Table XXII.

U. S. S. *TRENTON*.—CIRCULATING PUMP.

Revolutions per minute.	I.H.P.	Pressure in water chest of condenser, pounds per square inch.	Dimensions of pumps.
40	...	.10	Number of cylinders, 2
70	...	.20	Diameter " 11″
90	1.66	...	Stroke of piston, . . 9″
100	1.97	.35	Number of fans, . . . 2
110	2.39	...	Diameter " . . . 22″
120	2.82	...	Breadth of opening, 3″
130	3.28	.60	The fans deliver into a
140	3.80	...	12″ pipe entirely out-
160	4.96	...	side of the discs.
165	5.30	.90	
180	6.36	...	
195	7.56	1.30	
200	8.00		
220	9.89		
240	12.06		
260	14.39		
280	16.70		
300	18.93		
320	20.98		
340	22.35		
360	24.55		

Column 6—Feed Pumps.—The feed pumps are double and vertical, with shaft, connecting rods and fly-wheels. The diameter of the steam cylinder is 11 inches, of the water cylinders, 6 inches, and the stroke 10 inches. A great number of indicator cards were taken from one pump, which gave such discordant results that the horse powers could not be depended upon as those actually expended upon the feed.

Let D be the diameter of the steam cylinder, d the diameter of the water cylinder, p the steam pressure in the boilers,

and p_m the mean pressure in the steam cylinder; then p_m must exceed $p \frac{d^2}{D^2}$, if the pump cylinders be kept entirely full of water. As a matter of fact, the pumps are never entirely full, and p_m falls below $p \frac{d^2}{D^2}$, sometimes far below it. The mean steam pressure varied from 18 to 32 pounds, while the revolutions varied from 27 to 80, and the lower mean pressure often accompanied the higher speed.

I give below a table of the data taken from a few cards. The average of all the pressures is taken as the mean pressure on the steam piston. One pump is then supposed to run at a speed that will supply the boilers with water, assuming the efficiency at .5, and the indicated horse power is calculated on that basis. The terminal pressure taken to estimate the water expended on the engine is the average of all the cards.

Table XXIII.

DATA FROM ONE FEED PUMP.

Revs. per minute.	Mean steam pressure.	Terminal pressure.	I.H.P. at max. speed.	Water per I.H.P.
27	22.61	25.8		
58	18.23	23.5		
80	23.37	29.5		
65	31.97	36.3		
	24.05	28.8	28.16	44.8

Weight of water per I.H.P. per hour = 51.52.

The useful work expended in feeding all the boilers at maximum speed is 15.71 horse power, the volume of water 40 cubic feet per minute and the revolutions of the pump 122 per minute. Under banked fires, a constant feed is required to supply losses, the dynamo and distiller, but as the pump is run intermittently, the power does not amount to more than 1 I.H.P. I have taken it at 2, however, as the steam is used very wastefully.

Let H be any power of the main engine, H_m the maximum power; then the I.H.P. required for the feed pump will be $2 + (28 - 2)\frac{H}{H_m}$, or $2 + H \times .00412$. The values given in column 6 have been obtained from this formula.

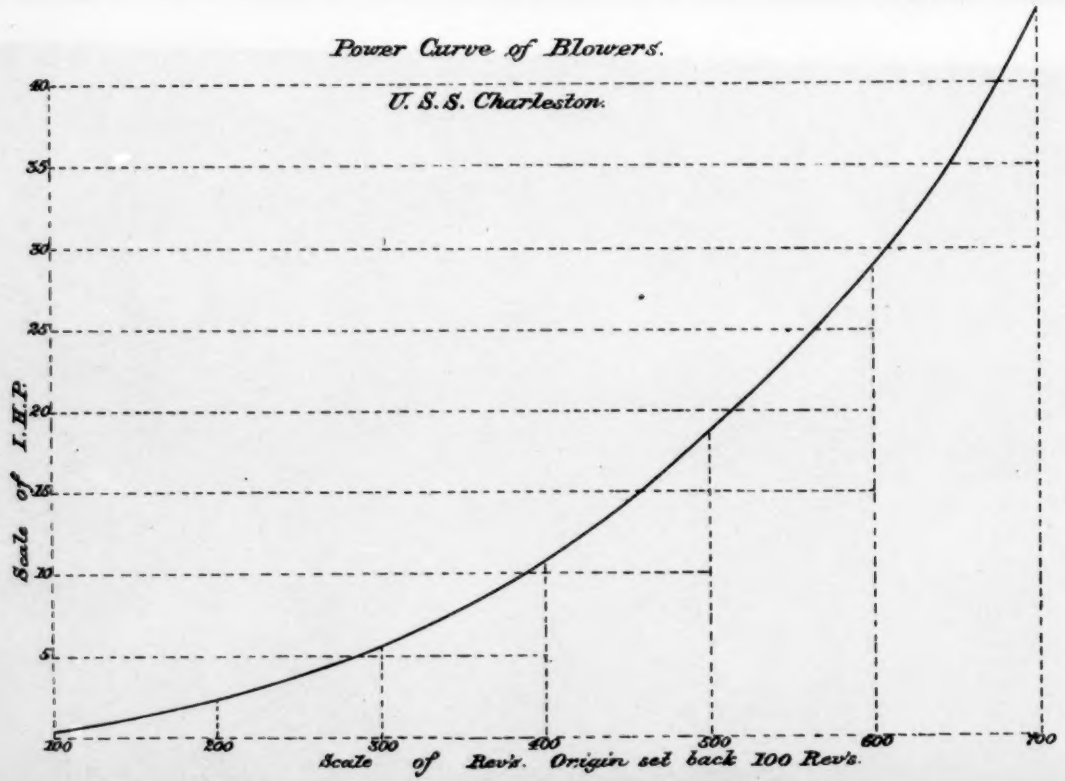
Column 7—Blowers.—The blowers are driven by simple horizontal engines of 9 inches diameter by 6 inches stroke. Each fan has a diameter of 60 inches by $7\frac{1}{2}$ inches breadth of opening. There is very little expansion of steam. A number of cards were taken at revolutions varying from 90 up to 692 per minute, and curves of mean pressures and horse powers were plotted to correct the results. I give both a table of the indicated powers and the curve. The steam cylinder from which the cards were taken was located in a very hot place where it was extremely difficult to work, so that I cannot claim great accuracy for the results. The horse powers vary nearly as the squares of the revolutions below 100, increasing to the cubes above 600.

Table XXIV.

DATA FOR ONE BLOWER.

Revolutions per minute.	I.H.P.
692	42.67
632	33.06
504	18.95
467	15.80
402	10.90
379	9.49
292	5.20
236	3.18
191	2.02
90	.76

The blowers were never needed with all boilers until the speed passed 13 knots. The revolutions are, therefore, proportional to excess of main-engine power above 2,220. Let R be the revolutions of the blower, R_m the revolutions for 18 knots speed, and H the indicated horse power of the main engines; then (taking $R_m = 486$ at 18 knots speed and 6,120 I.H.P.),


Power Curve of Blowers.
U. S. S. Charleston.
40
35
30
25
20
15
10
5
Scale of I.H.P.
100
200
300
400
500
600
700
Scale of Rev's. Origin set back 100 Rev's.

$$R = (H - 2220)\frac{486}{6120 - 2220} = \frac{H}{8} - 278.$$

The horse powers given in column 7 are simply measured on the curve after having determined the revolutions, bearing in mind that there are four blowers. The water expended is about 57.04 pounds per I.H.P. per hour.

Column 8—Main Bilge Pumps.—These pumps are constantly running and the power does not vary much. Cards were taken on the trial trip from which the horse power set down is reckoned.

Column 9—Auxiliaries.—A few auxiliaries, besides those necessary for the main engines, are constantly running at powers which do not vary with different speeds. I have, therefore, taken them exactly as determined on the trial trip, as follows: Steering engines, 3 I.H.P.; ventilating fans, 5 I.H.P.; flushing pump, 5 I.H.P.; dynamo engine, 13.5 I.H.P. No allowance is made for the accumulator, which is very wasteful of steam.

Column 10.—This column is simply the summation of all auxiliaries. It will be seen how little the work of auxiliaries varies from 4 up to 11 knots.

Column 11.—The coal expended for the auxiliaries per horse power per hour is taken at 6 pounds, which is probably rather low. It is based on an evaporation of 9 pounds of water per pound of coal.

Column 12 explains itself.

Column 13.—The coal expended on radiation, steam and water leaks from boilers and steam pipes is based on the weight of coal required for banked fires when no machinery is running.

This is an absolute loss which is constantly going on at low and high speeds. Its amount depends upon the number of boilers in use, and for the *Charleston* is 125 pounds per hour for each boiler. This column makes clear why the lowest number of boilers possible for a given speed should always be used, and why the additional boilers required when keeping distance in fleet steaming are a cause of loss. In the case of the *Charleston*, I have considered the lower speeds made with three boilers, although 4, 5 and 6 knots could have been made with two. The saving

in the latter case would have enabled the ship to make 5.84, 6.43 and 6.72 knots on one ton of coal.

Column 14.—The expenditure for distilling is taken at the average for a long run considering that 700 gallons of water are used per day; the fire pump is run for two hours; and the heaters are put on very moderately.

Column 16 gives the total coal for all purposes that can be estimated with any degree of accuracy.

The other columns explain themselves. In column 17, the coal expenditure was found to be somewhat lower than that given in the log. At 8.77 knots, for instance, the amount used would be 29.25 by the table, but the record of a long voyage shows it to be 32.25, a difference of 3 tons. While this loss must vary according to some function of the speed, its value is very difficult to obtain without a series of careful observations. An examination of the steam log shows at speeds below 10 knots some increase as the speed decreases, and at 5 knots it would probably be nearer 4 tons than 3. I have taken it as sensibly constant. Below 10 knots it will be found so, and above that speed its value makes no difference in determining the most economical speed. It is made up of all the losses not previously accounted for; such as general effect of weather, sea, draught, displacement, condition of boiler tubes and leakage by the packing rings. One case on the *Charleston* will show how great the last may be. The steering engine has a steam accumulator, and there is a leak past the packing ring. The under side of the steam piston is in continuous communication with the condenser, through a 1-inch valve and pipe, but the pressure is never less than atmospheric. I have taken it at 15 pounds and estimated the loss of steam through a 1-inch opening. It cannot be less than 840 pounds per hour, or one ton of coal a day. This leak is the same at all rates of running. The leak past the piston valves and main piston is probably greater at lower speeds.

The economical speed is near 8 knots. The results are not strictly accurate, but the table will demonstrate how much might be gained by a thorough test of the auxiliaries on board some of the new ships.

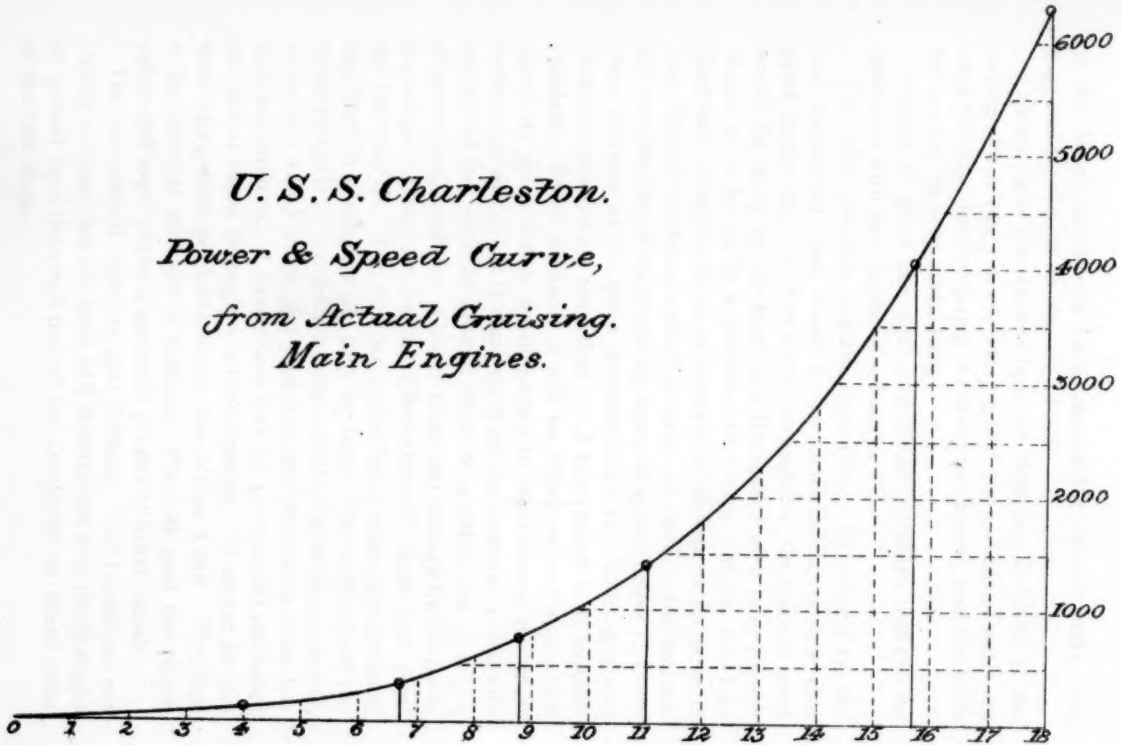


TABLE SHOWING EXPENDITURES OF COAL FOR U. S. S. *CHARLESTON*

TABLE SHOWING EXPENDITURES OF COAL FOR U. S. S. *CHARLESTON*.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Knots speed.	I.H.P. main engines.	Coal per I.H.P.	I.H.P. air pump.	I.H.P. circ. pump.	I.H.P. feed pump.	I.H.P. blower.	I.H.P. main bilge pump.	I.H.P. other auxiliaries.	I.H.P. total auxiliaries.	Coal per I.H.P. auxs.	Coal per hour auxs.	Coal for radiation, &c.	Coal for distilling, &c.	Coal for main engines.	Coal, total per hour. lbs.	Coal, tons per day.	Coal, tons per day. Increased 3 tons.	Knots per ton of coal.	Percentage of coal expended on main engines.
4	107	4.0	23.7	19.4	2.5	0.	6.9	26.5	79.0	6	474	375	100	428	1377	14.75	17.75	5.41	31.1
5	180	3.5	23.9	20.1	2.8	0.	6.9	26.5	80.2	6	481	375	100	630	1586	16.99	19.99	6.00	39.8
6	270	3.25	24.2	20.8	3.1	0.	6.9	26.5	81.5	6	489	375	100	878	1842	19.76	22.76	6.33	47.3
7	380	3.0	24.6	21.7	3.6	0.	6.9	26.5	83.3	6	499	375	100	1140	2114	22.65	25.65	6.55	53.9
8	520	2.8	25.1	23.2	4.2	0.	6.9	26.5	85.9	6	515	375	100	1456	2446	26.22	29.22	6.57	59.5
9	700	2.6	25.7	24.6	5.0	0.	6.9	26.5	88.7	6	532	375	100	1820	2827	30.30	33.30	6.48	64.4
10	960	2.5	26.5	27.3	6.2	0.	6.9	26.5	93.4	6	560	500	100	2400	3560	38.13	41.13	5.83	67.4
11	1290	2.4	27.6	30.6	7.5	0.	6.9	26.5	99.1	6	595	500	100	3096	4291	45.96	48.96	5.39	72.1
12	1710	2.3	29.0	35.9	9.3	0.	6.9	26.5	107.6	6	646	625	100	3933	5304	56.79	59.79	4.82	74.1
13	2220	2.2	30.7	41.1	11.4	0.	6.9	26.5	116.6	6	700	750	100	4884	6434	68.94	71.94	4.34	75.9
14	2820	2.1	32.7	47.5	14.0	2.5	6.9	26.5	130.1	6	781	750	100	5922	7553	80.94	83.94	4.00	78.3
15	3550	2.0	35.2	58.0	17.1	6.4	6.9	26.5	150.1	6	901	750	100	7100	8851	94.83	97.83	3.68	80.2
16	4370	1.9	37.9	70.0	20.6	17.6	6.9	26.5	179.5	6	1077	750	100	8303	10230	109.62	112.62	3.41	81.2
17	5220	2.1	40.7	84.8	24.0	36.8	6.9	26.5	219.7	6	1318	750	100	10962	13130	140.68	143.68	2.84	83.5
18	6120	2.5	43.7	114.0	28.0	69.6	6.9	26.5	288.7	6	1732	750	100	15300	17882	191.60	194.60	2.22	85.5

LECTURE SECOND.

Boilers and their Deterioration.

In the introduction to a chapter on boilers, Mr. Sennett says: "The efficiency of a war ship in the present day may be measured mainly, if not wholly, by the efficiency of her machinery, so that if, by mismanagement or want of care, the boilers are injured, the evil effect cannot be estimated by the depreciation in the value of the boilers alone, which would be comparatively a small matter, but it must be measured by the fact that the efficiency of the ship for the purpose for which she is designed is decreased, which is a much more important consideration. Necessarily, the greater part of the work connected with the care and preservation of the boilers is professional, and mainly, if not entirely, lies in the province of the engineer; but many points, particularly those relating to the management of the boilers when in active service, are, to a very great extent, under the control of the captain, and the efficiency and durability of the boilers will depend greatly on his knowledge of the subject."

The above is eminently true of our own Navy, particularly during the transition period, when every officer should learn how to take care of the ships we have succeeded in getting.

There is no other part of a cruiser so important as the boilers. The guns, torpedoes and main engines take care of themselves with a very moderate amount of attention, but the boilers are unremitting in their cries for help. When a ship is laid up, it is the boilers in nine cases out of ten. Some years ago I asked an old officer what a man's career in the Navy depended upon. He said: "Not brains, but stomach. The man with a sound stomach has only to wait until the man with brains wears himself out." So I might say of a ship. In the long run she will depend for her success upon the boilers. I shall, therefore, make no apology for dwelling at length on various questions which have been talked about for thirty years. Every commanding officer should know this subject of all others, and upon his

appreciation of it will depend, to a great extent, the interest and care taken by the engineer. The abstract principle of duty will not hold an officer to his work if he knows that his commanding officer does not understand or appreciate what he is doing, and is, besides, utterly ignorant of the care and thought required to keep machinery and boilers in a high state of efficiency. This subject divides itself naturally into two parts; first, the ailments which boilers are subject to and their cure; second, the modern improvements tending to increase the life and efficiency.

SCALE.

Almost all of the ailments have their origin in the use of sea water, which contains substances very small in proportion to the water itself, but which, in the long run, do the greatest damage. The first and most vexatious thing an engineer has to deal with is scale, or incrustation. It has many constituents, but there are three which so nearly make up the bulk of ordinary deposits that the others may be neglected. Salt itself, does no damage, nor does any other soluble compound which has no action on iron. Hence, if those substances which are deposited could be rendered soluble by any process, or be converted into harmless compounds, the whole question of boiler scale would be solved. Let me state the three ingredients mentioned above and their properties.

1st. "Calcic carbonate is practically insoluble in pure water; but carbonic acid, which is present in all natural waters, when in contact with calcic carbonate, converts it into bicarbonate, which is soluble. This applies equally to sea water."

If the carbonic acid is driven off by heat, the calcic carbonate falls at once as a fine powder and is deposited on the braces, tubes, and shell of the boiler. This deposit may be seen at any time upon removing the man-hole plates, as a white flaky powder, if the sea water be free from mud; or a brownish powder, if muddy, or action has taken place on the metal of the boiler. This compound does no harm, and one of the favorite remedies for scale is to convert the harder constituents into the carbonate by the use of chemicals which are introduced with the feed water.

The carbonate can be washed out by the fire hose at the end of every run. Its only fault is the injury to valves which may have their seats cut, or the outlets partly closed.

2d. Magnesic chloride in sea water becomes magnesic hydrate in the scale. Chemists do not agree about the reaction which takes place within the boiler. Many ascribe the corrosion and pitting in part to the hydrochloric acid which escapes, but there is no evidence in the shape of chloride of iron which would have to be found in the water of the boiler if such a change took place. The generally accepted theory is that magnesic chloride and calcic carbonate act upon each other in the presence of heat and form soluble salts in connection with the deposit of magnesic hydrate; this explains why so little calcic carbonate is found in boiler deposit. However, this compound exists in such minute quantities that it is of no great importance.

3d. Calcic sulphate is the most vexatious constituent of sea water. The frequent overheating and collapsing of furnaces, the burning of tube plates and the leaking of tube ends are mainly due to this deposit. One of the most interesting periods of the *Charleston's* pursuit of the *Itata* occurred at Acapulco where we missed her about 6 hours. She steamed directly south on a meridian for 150 miles and stopped 50 hours to clean a boiler and repair a bulged crown sheet. Some scale, consisting mainly of the above salt, had cracked from the tubes and dropped on the furnace crown where it had baked in with enough carbonate to form a thick mass, which kept the water from the metal and allowed it to become red hot. In the meantime, the *Charleston* had coaled in Acapulco and passed fifty miles to the eastward on her way to Callao. This only illustrates what has happened to many of our own ships, and may happen again whenever boilers are run for a long period using sea water without taking time to clean.

" Calcic sulphate is dissolved in water by the solvent power of the water itself, and not through the agency of carbon dioxide, and, therefore, merely boiling water at ordinary pressures does not suffice to cause its deposition. It is more soluble in a saline solution, such as sea water, than fresh water, but its solubility

rapidly decreases, (1) on the concentration of the saline solution, and (2) on the increase of temperature and pressure. The deposition of calcic sulphate during the concentration of sea water is, therefore, due to two causes; in the first place, although it is more soluble in a dilute solution of salt than in fresh water, yet it reaches its maximum solubility at a density of 1.033, and after this point concentration of the saline solution diminishes the amount which can be held in solution, and calcic sulphate is perfectly insoluble in a saturated brine. The second cause for its deposition is the rise in boiling point which accompanies the increase of pressure; the higher the temperature, the less the solubility of the calcic sulphate becomes, so that when 280° to 302° Fahrenheit is reached it becomes perfectly insoluble, both in sea and fresh water."

Here we see at once why the actual limit of saturation carried in a boiler is not of much importance, providing the water does not become a saturated solution of common salt. After the sulphate is once deposited, which takes place at once in high temperatures without regard to density, no further harm can be done. The old idea, therefore, of keeping the point of saturation down to $1\frac{3}{4}$ was a mistake, and the lower limit of 4 adopted by the Bureau of Steam Engineering agrees much better with the last ten years' experience. Besides, the higher the saturation, the less sea water for feed, hence the less sulphate and loss from blowing off. The deposit from sulphate becomes very important with the high pressure boilers now in use. If any calcic sulphate exists at all, it will be deposited and form a hard, tough scale which can be removed only with a hammer and chisel bar. It settles amongst the tubes where it cannot be reached and makes sea water very difficult for the engineer to deal with, if not practically impossible. Hence, all the later merchant ships use fresh water, and have special distilling apparatus for supplying waste.

There is no necessity of going into this question at length, as all officers know something about it, and know, also, that this compound must be dealt with under many conditions of cruising. From my own experience, I might almost say that the part played

by calcic sulphate in a foreign war will be as important as gun powder. The cruiser which neglects it will be lying in port under repairs, or liable to break down in an emergency, while the more industrious ship will be up to her greatest efficiency. In addition to the danger of explosion from overheating or cracking of plates, is "the great loss and waste of fuel entailed by the presence of boiler scale. The latest estimates go to show that $\frac{1}{8}$ of an inch of scale necessitates the use of 16 per cent. more fuel, $\frac{1}{4}$ of an inch, 50 per cent., and $\frac{1}{2}$ an inch, 150 per cent. additional coal." Added to this is the corrosion beneath the scale resulting from overheating and breaking up of water into hydrogen and oxygen, which attacks iron very rapidly.

The importance of preventing the incrustation above described has given rise to a thousand and one inventions, every one of which has had its day like the ephemerid. "Every conceivable substance has been tried, and has failed, because it has either had an injurious effect upon the metal, or produced an enormous bulk of loose deposit which rapidly choked the boiler outlets and made matters even worse than with the scale." There are in general, two methods of treatment; (1) mechanical separation of parts and consequent softening of scale, which may be dismissed at once as of no use, (2) chemical treatment within or without the boiler. Any chemical used within the boiler may cause great deposits of soft scale which cannot be removed in ordinary cruising and which accumulate rapidly enough to cause the same damage that a thinner scale might cause.

The treatment outside of the boiler with sodic carbonate seems to promise the best results. It is said to have given fine results when used in tanks within which steam could be admitted to bring the pressure to 10 pounds above the atmosphere. I see no reason why two cylindrical tanks holding 1,000 gallons of water each could not be placed on opposite sides of a ship, and braced to stand 10 pounds pressure. All salt feed could be taken directly from them after heating with sodic carbonate and rapid filtration. These tanks could be made very useful for bringing the ship to an even keel upon arriving in port, and the deposits within them would be easy to get at for cleaning. At

present, there is no certain remedy for scale except hard and constant work inside of the boiler. There is a theory that the scale is dissolved by letting water stand in the boiler after arriving in port. This is to some extent true, especially if the water is slightly above sea density. At any rate, it softens the scale and has the additional good quality of checking corrosion, if a small quantity of caustic soda be added. The water level must be kept well above the normal level in order not to expose the same line to the action of air and water. All washing and scaling should be done immediately after the water is taken out, otherwise the carbonate mixes with sulphate and dries into a hard mass difficult to remove.

The true remedy for scale is the constant use of fresh water procured from shore and supplemented by an evaporator or distiller during the cruise. As the evaporators often give trouble, I am sure the addition of the above-mentioned tanks would be a great gain where salt feed has to be used. It may be of interest to know the average amount of water necessary "to make up the volume of condenser water to the amount required for the boilers. It is about one ton a day for 1,000 horse power, if the engines be in fair condition and reasonable precautions are taken to prevent waste." In this connection, let me state that sodic carbonate is in use at the Pinole Powder Works, where I have seen four or five boilers in excellent condition after years of service with well water even worse than sea water in deleterious elements.

I have stated a few rules at the end of the first part of the lecture for the guidance of commanding officers, which include the precautions against scale.

A scale sometimes follows the excessive use of cylinder oil which is deposited on heating surfaces and baked into a hard mass. There are cases of overheating due to this. Also a thick deposit and great wearing of valves are caused by the use of muddy river water.

CORROSION AND PITTING.

Corrosion and pitting are the most fruitful causes of decay in marine boilers, and their prevention has formed the subject of

many essays and books. The latest fad is galvanic action to which is ascribed everything from plain rust to gross carelessness. In an experience of eight years at sea, I have never seen a case of decay that could be ascribed unequivocally to the electric current. Such a current in sea water follows as the natural result of chemical change and is not the primary cause of the wasting away of metals. For many years after the introduction of the surface condenser the rapid decay of boilers was ascribed to a variety of causes, one after the other, but they have all had their day. At first, chemists and engineers laid the pitting of plates to animal oils used in the steam cylinders. When mineral oils were introduced and the pitting continued, they asserted that the distilled water acted on the brass condenser tubes and carried particles of copper over to the boilers where voltaic couples were formed and the boiler shell suffered in consequence. They forgot that, at that time, boiler tubes were made of brass, and it was not necessary to go out of the boiler for the guilty couple. They also discovered all sorts of organic salts in the water due to the breaking up of the oils. I think we have a clearer notion of what takes place now, although we have not succeeded in procuring the remedy.

CORROSION.

The main phenomena of boiler decay, as I stated above, are corrosion and pitting. The former may be put down as rusting caused by the presence of four substances, moisture, carbonic acid, oxygen and iron. As Prof. Lewes (whose book on Service Chemistry every officer should read, and from which I have taken several extracts in the above) says: "In the presence of moisture, carbonic acid and oxygen simultaneously attack the iron or steel forming a thin layer of ferrous carbonate. The ferrous carbonate is further acted on and ferric oxide and ferric hydrate are formed. During these reactions, the carbon dioxide is liberated in the moist metal, and reacting with more oxygen from the air, carries on the process of corrosion which is now further accelerated by the fact that the hydrated oxide on the surface of the metal is electro-negative to the metal itself, and, excited by the presence of the moisture and carbon dioxide,

creates a galvanic current at the expense of the iron, and the mass of rust formed being porous, the action continues until all the metal is destroyed and has returned to its natural condition of hydrated oxide." Sea water contains all the elements necessary to act upon iron, besides the saline constituents which tend to increase the rapidity of the change both by chemical action and the resulting current. Rusting is also naturally increased by want of homogeneity in the metal, by particles of rust, by mill scale and even by different metals in contact with each other. Experiment has shown that steel may rust faster than wrought iron, also that with both of them the corrosion takes place at an increasing rate; thus disposing of the old idea that rust protects iron. It may, if the iron is kept perfectly dry; in which case it does not need protection.

Very little corrosion takes place when a boiler is in use, as oxygen and carbonic acid are driven off with the steam. The most favorable conditions for decay in the Navy exist when the water is pumped out and the boilers often have to be left wet and exposed to the air.

Distilled water is also said to have a corrosive action on iron and steel, and it is probable that the feed water which is taken pure from the condenser does attack those parts of a boiler which are exposed. There is also no doubt that the breaking up of the cylinder oils, whether animal or mineral, liberates various compounds which, in company with sea water at a high temperature, attack the plates. It is also asserted that, at very high temperatures, pure water is decomposed by iron and black oxide is formed. The last three causes of decay go on when a boiler is in use, and we have no certain remedy for them.

PITTING.

Pitting is probably due to the same action as corrosion, concentrated on spots which are predisposed by certain faults and irregularities in the metal to decay. Spots of rust are formed and, perhaps, particles of foreign metals are carried in with the feed water, which hasten the action in localities. Also, the action once started, the galvanic current may help it on. Mill scale is said to form a couple with the iron or steel.

There are two kinds of remedies applied, one to treat the water outside of the boiler with some alkali which prevents chemical action and the other to introduce a substance into the boiler more readily attacked than iron. Usually, caustic soda or common washing soda is used for the former, and small quantities are introduced into the condenser at regular intervals. I think the preparation of salt water with sodic carbonate as described in connection with scale is probably as serviceable in checking corrosion and pitting as in reducing scale.

In the latter case, zinc slabs are placed inside of the boiler in baskets, with an attempt to secure metallic contact. It is said that the zinc forms the positive pole of a couple with the iron and wastes away, forming a solution of zinc salts in the water. There is a great amount of testimony both for and against the use of zinc. It is an expensive remedy, and a thorough test in one of our ships to establish beyond question its utility would be a very good thing. I think that it may be an efficient offset to carelessness and rough treatment in continuous service. Where a boiler is badly treated, zinc may help to check the injuries that might result. Its action in the sea water is little understood, but the weight of opinion is that galvanic action has nothing to do with its beneficial effects, except as accompanying the chemical action which really does take place. Hot brine is known to dissolve zinc, and a basic solution is probably formed which has exactly the same effect as caustic soda pumped in with the feed water; that is, it checks chemical action by forming compounds with the chemicals in the water which are not deleterious to iron.

Zinc may prevent local action to some extent by the prevention of currents, but it is introduced in quantities too small compared with the weight of the boiler to have any great effect except in solution. One fact in regard to this matter which seems to establish galvanic action, is that zinc is much more rapidly eaten away in a new boiler than in an old one whose surfaces are partly covered with non-conducting substances; but on the other hand, many engineers testify that the metallic contact, which is essential to the above current, is not required for the beneficial effect of zinc. Inasmuch as metallic contact does

not last more than twenty-four hours after getting up steam, they are probably right.

I quote the testimony of Mr. Walliker, of Lloyd's, on this subject. He says, "that a great part of his business consists in examining for corrosion and pitting and all concerning the condition of boilers. During the last seven years his observations have led him to conclude that 75 per cent. of the boilers trading from the Tyne use zinc, and 25 per cent. patent composition or nothing at all." His opinion about zinc is this, "that in ocean-going steamers, not subject to very bad water—water contaminated by sewage and other deteriorating matter—the use of zinc is not in any degree necessary, and is only a safeguard against careless engineering."

The tubes of our boilers, particularly the *Charleston* and *Yorktown*, have had to be replaced in three years, while the boiler shells have not deteriorated materially. We have been able to get at all the surfaces of the shells and furnaces, and totally unable to clean the tubes on account of inaccessibility. Their thinness has also made a very slight amount of corrosion apparent. The *Charleston's* tubes pitted very badly from the start, and we could never tell when one of them was going to blow out. The holes occurred indifferently throughout the tubes, and I think are accounted for by the presence of scale partly cracked off, which held the moisture and gases in certain localities where they could attack the iron when the boiler was not in use. Additional zinc slabs were placed in the neighborhood of the tubes without any decrease in the action going on.

The Bureau of Steam Engineering has required all steel plates for boilers to be pickled in a dilute solution of hydrochloric acid for the removal of mill scale and other impurities.

HEAT EFFECTS.

The effects of heat undoubtedly shorten the life of a boiler, outside of all those considerations connected with hastening corrosion and chemical change within the boiler, hardening scale, burning furnace crowns and tubes beneath the scale, and other like changes. I refer to the changes of shape and wasting

away of metal in contact with the flame. When fires are started, the various parts of the boiler begin to change shape, the upper part of each furnace expands faster than the lower half and bends the furnace generally into a bow, the tubes lengthen and push the expanded ends a short distance through the tube sheet, especially near the flanges, which are rigid, also expanding in diameter more than the hole which they fill, and, as steam forms, the parts of the shell of the boiler expand, both lengthwise and in diameter, at different rates. These alterations gradually draw the rivets out of shape and thus cause leaky joints, bend the stay bolts, and start cracks in various parts of the boiler, which grow more or less rapidly according to the treatment the boiler receives. They frequently end in disaster. Engineers know that tubes often leak under banked fires, for the reason that they can not recover their original shape after the change described above, both because the shell is kept hot by steam while the tubes are allowed to cool down to the temperature of the surrounding water, and the pressure against all the internal parts is kept on. When cold, a boiler which has been steaming for some time will almost always leak around the tubes with the pressure on. Of course, the presence of any scale only increases the effect of the flame by making the differences in the temperatures greater. The tube ends are also wasted away by the flame which impinges directly upon them, particularly if the coal contains sulphur. The introduction of cold water is also extremely bad. The evil effects of heat are often increased by poor circulation of water. The English have assigned much of their trouble with tubes under forced draft to a formation of superheated steam on the water side of the tube sheet, which cannot flow off readily and allow the cooler water to keep the tube sheet down to the same temperature as the surrounding metals. This also applies to the tube ends.

It is here that the captain of a ship can do much to lengthen the life of a boiler and lessen the repairs. No sudden changes should ever be required. If steam is wanted, the chief engineer should be informed at least thirty-six hours beforehand, so that he can heat the water to boiling point by means of the hydro-

kineters before he even starts fires. The fires should then be started slowly and steam gradually worked up to the required pressure, taking at least three hours. Where ships are supplied with circulation pumps instead of hydrokineters, a longer time may be taken to get up steam, while a circulation is kept up by the auxiliary boiler. After the engines are once ready, there should be no delay necessitating opening of furnace and connection doors, putting up ash pans, or pumping cold water into the boiler. The cheapest way to control the steam pressure is to waste the steam through the safety valve. Uniform and steady steaming, such as merchant ships have, is the most favorable to the long life of a boiler.

The sudden changes spoken of above are brought about by banking fires, hauling fires, allowing cold air to strike the tubes, blowing down boilers, pumping and blowing to reduce saturation, and, lastly, the use of blowers which will be spoken of later.

The introduction of blowers, in order to increase manœuvring power and speed on emergency, is safe enough, if proper precautions are taken, remembering that forced draft renders all changes in the boilers greater and more sudden. With an air pressure in the fire room, every opening of fire doors for cleaning or coaling furnaces introduces a large volume of cold air into a very hot combustion chamber. It is always preferable to put on another boiler rather than to run the blowers with pressure above $\frac{1}{4}$ inch of water.

In this connection, I will say that a high smoke pipe is much the best method of securing good draft.

Whatever may be the reason, the draft induced by producing a partial vacuum in the smoke pipe is much less injurious to the tubes than any system of driving the air into the ash pans. With a lofty smoke pipe, there is no complication in the fire rooms, which are open and cool, and the combustion may be increased from 30 to 40 per cent. by simply adding 40 feet of height. As will readily be understood, the economy is thus increased by dispensing with the coal and men required to run the blowers, the life of the boiler lengthened, and the sustained sea speed brought up one or two knots, inasmuch as the ad-

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ditional coal can be burned without starting leaks in the combustion chamber and choking up the tubes with salt. While there is absolutely no objection to a high smoke pipe from an engineering standpoint, I do not think there is any objection for war purposes that cannot be urged against any smoke pipe. It is certainly equivalent to the addition of one or more boilers. The question of sightliness amounts to nothing, as a war ship is built solely to fight her guns and torpedoes or to ram. A trial of a smoke pipe 120 feet high by the Denny Bros. in the merchant service has given very satisfactory results. I do not advocate doing away with blowers for manœuvring and reserve power; but I do think they should be kept for such duty. As stated before, the blowers form ready means of bringing up the speed two or three knots within a few minutes, and their adoption is a necessity for war ships.

RULES FOR TREATMENT OF BOILERS.

The following rules strike me as those most essential in the care of boilers. I have tried to make them a *rêsumé* of the experience of engineers generally, and I give them also as the result of my own observations at sea, which further experience may change:

1st. Use fresh water from shore whenever it can be obtained, even if expensive, and carry a supply in a double bottom, if the ship needs ballast. There will be considerable saving in coal where the run is to be a long one, inasmuch as a thinner scale is deposited, and blowing for saturation will not be necessary, added to the increase in the life of the boiler and the lessening of work for the firemen.

2d. Use the evaporator for replacing all water, if it can be used without entailing too much work in cleaning out salt from the coils.

3d. Use the salt feed as little as possible in order not to carry air and calcic sulphate into the boiler, even if it be necessary to let the saturation run up to five or six.

4th. Reduce the quantity of cylinder oil to the lowest possible limit. Much greater watchfulness will have to be exercised on

the main engines to insure that no cutting takes place, but the deleterious action of acids from the breaking up of oil will be reduced to a minimum.

5th. Inject a small quantity of caustic soda dissolved in fresh water (enough to saponify the cylinder oils) into the condensers once or twice a watch, in order to render the water in the boiler neutral to the iron.

6th. Avoid too frequent blowing off, in order not to waste the water. A slight blow from the surface to remove the oily scum which forms in every boiler will be found sufficient once or twice a day.

7th. Work the boiler with regularity. Frequent changes of temperatures unquestionably hasten any action which has begun.

8th. Clean the boilers with regularity, even if one boiler has to be left open at sea and a working gang taken out of the watch list. This point I consider next to the use of fresh water in importance. The captain of a ship should take a greater pride in having clean boilers than in having a clean deck.

9th. In port, let the fires die out, pump out the boiler when cold and give great attention to cleaning furnaces, tubes and inside surfaces; then fill the boiler above steaming level with fresh water in which caustic soda has been introduced, bring the water to a boiling point with the hydrokineters or a light fire, while the air cocks are kept open for the escape of gases. In case there is not time to clean the inside surfaces, do not take the water out at all, if fresh, or until the day before getting steam, if salt.

10th. Keep a considerable weight of zinc well distributed around the surfaces below the water level. I add this not from real faith in the good effects of the metal, but out of deference to the judgment of many engineers who no doubt have good reasons for their belief in it.

11th. Never use the boilers for trimming ship.

12th. On the first voyage of new boilers, use enough sea water to deposit a thin hard scale on the surfaces. (For a protection against corrosion, if sea water is to be used afterwards.)

13th. When at sea, never carry salt water in cold boilers unless they are entirely filled, as the swashing about of sea water in contact with air will cut away the water line quite rapidly.

14th. Take an intelligent interest in what goes on in the fire room, and make the care of the boilers the first consideration at all times.

I cannot speak too strongly of these matters, and no commanding officer should neglect them. Questions of the rate of steaming, banked fires, and many other datails affecting the boilers are entirely within his control, except in stress of emergency, and there is no division of responsibility in case of disaster or large repair bill. Some of the above rules may seem trivial, but their violation is like the drunkard's next glass.

I come now to a subject which has aroused the attention of engineers generally; that is, the shortness of the life of Navy boilers when compared with merchant ship boilers. Some of it is due to irregularities of cruising incident to the service and can not be avoided. The responsibility for the rest of it may be divided equally between the commanding officers and chief engineers. It frequently happens that all the work an engineer may put into a boiler is thrown away by the utter carelessness of his commanding officer; again, it may happen that the chief engineer wants to get through the cruise as easily as possible without much thought about the next cruise.

Better results might be obtained if the officers of a ship remained longer with her, thus fixing responsibility more clearly, although I do not think that essential, if officers are conscientious in the performance of duty. But by far the greatest amount of deterioration arises from boilers lying idle for long periods, often empty and damp, and from too much irregularity in steaming. These two things, alone, I think, account for three-fourths of our trouble. How far they can be avoided I must leave to the judgment of commanding officers and engineers. The trans-atlantic liners use fresh water as much as possible, are thoroughly cleaned at the end of every passage, and steam at a constant rate from port to port. The boilers are never filled with salt water to trim ship. The life may be placed at fifteen

years as against ten to twelve for the Navy. We have a case of remarkable longevity in the boilers of the *Michigan*, built in 1859 and only now being replaced. This shows what fresh water will do, as her entire service has been on the Lakes.

CORROSION GENERALLY.

Before going to the construction of boilers, something may well be said on the subject of corrosion generally. The outside of the boiler shell, the supports of engines and boilers, the inner skin and iron pipes, are all liable to rust, and, in some cases, to pit. The former is due to moisture, oxygen, and carbonic acid, as stated before, and the only remedy is frequent cleaning and painting. Cement should be supplied on the inner skin from one to two inches thick in places that cannot be reached readily during the cruise, also to fill all depressions where water collects to the level of the limber holes for good drainage to the wells. The plates must be well cleaned and the cement carefully applied, so that no moisture can get beneath it during a cruise. This matter of good drainage is highly important, in order that the bilges may be kept dry and well painted. The inner skin, underneath the boilers, of the *Charleston* was very difficult to protect, as it could never be pumped dry. Small drain pipes leading into the bilges heated the water enough to soften the paint and render frequent scraping necessary. In three years corrosion had set in quite perceptibly. Pitting, though frequently ascribed to galvanic action, is probably due, in great measure, to acids formed from soot, coal dust and ashes, which get into the bilges from the fire rooms. It does not require much imagination to picture the decay which takes place with the above-mentioned materials, and two or three inches of water swashing back and forth under the boilers as the ship rolls. Wherever the paint has peeled or blistered corrosion must begin.

I made it my business to examine the inner skin of the *Charleston* during a period of two years and a half for evidences of galvanic action, and although I saw many places where the plates had corroded, I did not see one that could not be accounted for by plain rust or chemical action. The manhole plates into the

double bottoms were made of brass, and worked nearly flush with the bottom, but I could not discover that the plates were any worse in their neighborhood than in places remote from them. I do not wish to be understood to advocate the indiscriminate use of copper, brass and iron together where sea water may possibly form a battery. There is no more use in tempting Providence than there is in giving Greek names to things we have been familiar with from childhood. The outboard valves of the *Charleston* were all of bronze, and secured to the outer skin without zinc protecting strips. I could never find any pitting in their neighborhood, but I could find decided corrosion and pitting in the wake of the ash chute just below the water line and abaft the bilge discharge in the wake of bilge water and coal dust, which undoubtedly went out together.

Iron pipes, in which bilge and sea water are used, will usually give trouble within a cruise and have to be removed. It is a curious fact not yet explained, that the copper pipes in our new ships have also pitted badly wherever sea or bilge water is used. We often had to drive pine plugs into holes eaten through the bilge discharges of the *Charleston*. The main injection pipe of the *Baltimore* was so badly corroded within two and a half years from date of commission that it had to be taken out and a new one put in.

Although connected with the subject only in a general way, I will give a brief history here of a case of corrosion on the *Charleston* for the useful lessons which may be learned from it. On the passage from Yokohama to Honolulu, last December, the sea was unusually heavy for eight or nine days, with a strong wind on the port beam. The ship rolled more than usual—about eighteen degrees each way at times. We discovered one afternoon that she began to roll heavily to port, and that the difference between the two sides increased rapidly. Captain Remey first noticed it in the cabin, and we went about searching for the cause. The natural supposition was that the wing bunkers were filling, and all the coal chutes that could be reached were examined to see if water was coming in, but there was only an occasional small stream through the joints. We tried all the

bunker drain cocks underneath the protective deck, and found no water coming through. Assistant Engineer Jones stopped the flushing pump, with an idea that the water pipe in the middle wing bunker on the port side, leading to the closets, had given out, and that the drain cock had choked up with coal dust. This proved to be the trouble. All the water-tight doors in the bunkers had been closed, excepting in those which were empty, and we had to shovel the coal away to get at the closed doors. The bunker referred to above was found full to the berth deck, and water was running through the chute into the midship bunker, and from there into the port after-feeding bunker. We found a spot about one inch in diameter completely corroded through the flushing pipe, which passed through all the port bunkers, and was secured to the berth deck overhead. This pipe was of iron, and from one to two inches in diameter. The flushing pump had simply been filling up the ship. While the bunker doors had proven themselves tight, the drain cocks had been of no use. Some of them had to be taken off entirely to get rid of the water which flowed over after the bunker doors had been opened. The ship had a list of from seven to nine degrees before we stopped the pump, and that with one port bunker above the protective deck filled completely with coal and water, and the port after feeding bunker half full of water, in addition to the coal. Our position might have been unenviable if the water tight doors had not been closed when coaling ship. In addition to an object lesson on corrosion, the occurrence furnished food for thought to a commanding officer as to the method of fighting the guns of a ship with a few degrees list. The penetration of the hull near the water line might easily admit water enough to list the ship seven degrees, and bring the edge of the protective deck above water on the high side.

MECHANICAL CONSTRUCTION.

The types of boilers in common use are so well understood by the Navy generally, that I will not take the time to describe them, but will state only the advantages and disadvantages of each and the changes brought about by the demand for higher speeds. The ordinary marine boiler is return fire tubular, either double

or single end, containing from two to eight furnaces. It is well adapted to all war ships with depth enough to get the crown well below the water line. The flame impinges first on the back of the combustion chamber and turns into the tubes, which is a decided advantage. Its weight is not excessive, and with good forced draft, fully 2,500 horse power can be obtained from one boiler. Compare this with the *Trenton's* eight boilers for the same power. The parts can all be made accessible for cleaning and repairs.

The gunboat boiler is the only other boiler adapted for cruising ships. As its name indicates, it is used for the smaller ships with light draught. It is direct fire tubular and the flame impinges directly on the tube ends as in a locomotive. The diameter may be made considerably smaller than that of a return tubular boiler, for obvious reasons, but it occupies more space for the same power and is probably a little heavier. The *Charleston* is the only one of the larger ships fitted with these boilers, and I do not think they are well adapted to her. Double-end boilers, with two furnaces in each end and two smoke pipes, would give more steam and better cruising speed. The one point which distinguishes these boilers from others is the large steam surface, making it possible to force them to any extent without danger of lifting water.

The locomotive and tubular boilers cannot be successfully used with sea water, and are, therefore, never put on a cruising ship. As a reserve power for coast-defense ships, and in torpedo boats, they do very well. I will speak more fully on this subject later on.

In the matter of efficiency and improvements, the Navy is fully abreast of the times. The Bureau of Steam Engineering has been very conservative in reducing weights in the fire room; wisely, I think. The universal complaints in the English Navy about the boilers give enough proof of this. They have had one breakdown after another, while we have yet the first failure to record. Our ships have gone about their work without unnecessary delays and repairs, and the policy of Engineer-in-Chief Melville is fully justified.

WEIGHT OF BOILERS.

Name.	1 Number of boilers.	2 I.H.P. maximum speed.	3 Weight of boilers per maximum I.H.P., lbs.	4 I.H.P. sustained sea speed.	5 Weight per I.H.P. sustained speed., lbs.	6 Grate surface, sq. ft.	7 Weight per square foot grate surface, lbs.	8 Heating surface, sq. ft.	9 Weight per square foot heating surface, lbs.	10 Steam pressure, lbs.	11 Factor of safety for shell.	12 Weight of water in boilers per I.H.P. sust'ned speed, lbs.	13 Weight of boiler, water and appurtenances, per I.H.P. sustained speed, lbs.
Trenton	8	4,828	132	2,414	264	492	1,294	11,895	54.0	80	5	113	442
Concord	4	3,405	72	1,703	144	220	1,114	8,210	30.0	160	3.90	80	288
Charleston	6	6,667	67	3,333	134	421	1,060	15,147	29 0	90	4.8	74	254
San Francisco	4	9,913	54	4,956	108	553	972	19,658	27.0	135	4.40	57	196
New York	6	16,000	66	8,000	132	988	1,065	31,005	34.0	160	4.40	70	238
Columbia	8	20,500	67	10,250	135	1,344	1,030	43,268	32.0	160	4.40	75	247
Monterey	6	5,400	44	2,700	89	388	618	12,840	19.0	170	.....	.....	147
" Cylindrical boilers.	2	1,800	59	900	117	88	1,200	2,505	42.0	160	.....	57	.....
" Coil "	4	3,600	37	1,800	74	300	447	10,335	13.0	170	.....	11.2	.....
Cushing	2	1,875	20	940	40	76.6	497	4,750	8.0	250	.....	8.4	59

I have made out the accompanying table from the returns of weights at the different ship yards to show that the boiler scantling has been reduced as the improvements in materials have made it safe. The *Trenton* is taken as a type of the old Navy in order to compare the new with the old. In columns 2 and 3 are given the maximum indicated horse power and weight of boilers per horse power without appurtenances or water. A supposititious horse power is added for the *Trenton* to show what might have been the gain with the same machinery, but with blowers and a finer pitched screw. These two columns do not give a fair comparison, as the reduction of weight per horse power indicated is all due to forced draft and not to actual saving of materials. For this reason, columns 4 and 5, reckoned from the sustained horse power, are added. I think the best method of comparison is based on the grate surface, or the capacity for burning coal, although it does not take into consideration the efficiency of the heating surface. As will be seen, the weight per square foot of grate varies inversely as the tensile strength of the material. The other columns are added for general information. Reckoning in water and all appurtenances necessary to run the boilers, we get from 9 to 10 indicated horse power for one ton's weight in the fire rooms, when the boilers are worked under natural draft.

The *Monterey* has two cylindrical marine boilers supplemented by four Ward boilers, and they are separated for comparison. An enormous gain is shown to result from the employment of coil boilers here, but its practicability remains to be shown. The *Cushing*, with Thornycroft boiler, gives about the same results as the Ward boilers in the *Monterey*.

The table reveals a substantial gain resulting from the following changes in design and materials:

1st. Improved materials. The tests applied to steel are now so rigid and the methods of manufacture so much better that we are able to get steel of a far higher quality than even the Advisory Board could obtain ten years ago. It is vastly superior in tensile strength to the wrought iron used twenty years ago, and much more reliable. The reduction in percentage of phos-

phorus has also made it well suited to fire surfaces. In fact, it may be said that anything can now be done with steel.

2d. *Larger Dimensions of Plates.*—This alone has enabled the engineer to carry higher pressures and reduce the factor of safety from 7 in the old boilers to $4\frac{1}{2}$ for the new ships. Plates up to $1\frac{1}{32}$ inches thick have been successfully worked for the *Columbia* and *New York*, and the joints reduced to a minimum. The longitudinal joints are also treble riveted, giving 85 per cent. of the strength of the solid plate as against 70 per cent. in the *Trenton's* boilers.

3d. Furnaces are now made in one piece welded longitudinally.

4th. The introduction of corrugated furnaces has done much to increase pressures. As is well known, furnaces cannot be made thicker than $\frac{9}{16}$ inch without danger of cracking under heat, and some method of increasing the strength to resist collapse has to be adopted. The Adamson ring had its day. There is nothing better than the corrugated furnace, both for strength and adaptation to changes of shape under heat, its only disadvantage being liability to crack in the grooves, where scale sometimes lodges.

5th. The use of iron instead of brass tubes is really only following merchant-ship practice. It must be cheaper, although the life of iron tubes is comparatively short. We have renewed the tubes of both the *Charleston* and *Yorktown* after three years' use. Whether, in the long run, iron tubes will prove the best for naval purposes only experience will show. I do not think brass tubes could be used with forced draft and rigid boiler construction, on account of their great expansion under heat.

6th. The more extensive use of screw stay tubes has rendered the tube sheets very rigid, perhaps too rigid for high temperature. There should be play enough to allow the whole sheet to "breathe" as the temperature of the tubes rises.

7th. Ferrules have been placed over the fire ends of tubes of *Gunboats 5* and *6*, *Cruisers 7* and *8*, and the *Monadnock*, to protect them from excessive heat. These ferrules, however, prevent further expansion, if leak occurs, and reduce the size of the tube.

8th. The top ends of the larger boilers have been rounded off, to reduce the number of longitudinal braces and render the upper parts more accessible.

9th. The use of blowers has doubled the power to be obtained from a given size of boiler. There are two methods of delivering air to the furnaces: One with a closed fire room and a pressure from 0 to 3 inches of water maintained by discharging air drawn from the ventilators directly over the heads of the men; the other to discharge it into air ducts leading to the ash pans, leaving the fire rooms open. They both have their advocates, but the Bureau practice is decidedly in favor of ash-pan draft. After two years and a half with the closed fire room on the *Charleston*, I unhesitatingly pronounce against it as "an invention of the evil one." It shuts the men up in a small hole, entirely out of control of the engineer, where they are easily demoralized, and renders communication between fire and engine rooms very troublesome when the sliding doors are closed from deck. Access to the fire rooms was impossible, without going through a door in the protective deck, which was too heavy for one man to handle easily, and down a ladder too hot to touch. The cloud of coal dust continually driving into the men's faces was most irritating. There is no doubt of its injury to the tubes and combustion chamber on account of the impossibility of regulating the air supply when the fire doors are open. As stated before, some of the *Charleston's* boilers were rendered almost useless for making steam after ten hours with the blowers, and through no fault in the construction, as the work had been carefully done. I crawled into all of the combustion chambers after the trial, and found about half of the tubes stopped up with salt. The boilers contain the enormous combustion chambers common to all the furnaces that have given so much trouble in the English Navy, a heritage we acquired with the designs purchased in England.

The closed ash pit has many advantages in ready communication between different parts of machinery space, ease of regulation of air supply and better treatment of the boilers. Each furnace door is fitted with a latch, which shuts off the air supply

as the door is opened for coaling or cleaning furnaces. Other points to be considered are the comfort and confidence of the men. The *San Francisco*, which is fitted with this system of draft, tried one boiler under full power with blower running at full speed for thirteen hours. After cooling down, only half a dozen tubes were found weeping slightly; the others were as good as ever. The two ships were built by the same contractors, and the boilers even by the same workmen, so that the test of the two systems was in every way a fair one. In the table it will be seen that the *San Francisco's* boilers have a very small weight per square foot of grate surface. The combustion is also far better with ash-pan draft, owing to more uniform distribution of air in the furnaces. I quote from a report of the trial of the *San Francisco*, by Passed Assistant Engineer Warburton, who was also on the trials of the *Charleston*: "The same coal was used on the *Charleston's* trial, with closed fire rooms, when great volumes of smoke issued from the smoke pipe, and the shower of cinders on the deck was very much greater than on the *San Francisco*. Again, the *Charleston's* uptakes and smoke pipe at the base were at times red hot, but in the *San Francisco*, though the combined area of the two smoke pipes is smaller than that of the *Charleston*, there was no overheating and the paint was not blistered. The fire rooms were comparatively comfortable, and free from the clouds of dust which pervade air-tight fire rooms." In spite of the obvious advantages of the closed ash pit, the difficulty of locating the blowers and getting a supply of air to them, also in arranging the water-tight bulkheads, leads to the adoption of the closed fire room in many cases. At best, however, either system is full of weariness and vexation. The Howden system is simply the ash-pan draft with means of heating the air before it goes into the furnaces.

Dampers for controlling the draft and steam were fitted on the *Charleston*, but they were not an unqualified success. A very short run under assisted draft warped them out of shape, and we were lucky if they stuck open instead of closed.

In most cases the number of smoke pipes has increased with the number of boilers. This is necessary in order not to build

uptakes half the length of the ship, and to confine each group of boilers to its own compartment. There is another advantage in that the size of the smoke pipe will not be entirely disproportionate to the grate surface at low powers.

TUBULOUS BOILERS.

The introduction of these boilers is due entirely to the demand for higher pressures and reduction of weights. The limit of ordinary marine boilers with the materials in use at present is about 180 pounds steam pressure, unless the size be very much reduced; therefore, for quadruple-expansion engines, which require at least 200 pounds to get the benefit from expansion in four cylinders, another form of boiler will be necessary. This engine would be of doubtful economy in the Navy, at our cruising speeds, and I do not think the triple-expansion engine will be superseded for many years to come.

The performance of locomotive and tubulous boilers is not entirely satisfactory. Many accidents have occurred on torpedo boats, although the damage done by an explosion is reduced to a minimum by confining its area to few tubes instead of a large boiler shell. They generate steam rapidly and can probably be forced more. The heating surface is usually far in excess of that of Scotch boilers for the same grate surface, and the parts are reasonably accessible for cleaning and repairs.

There is no necessity of a full description of these boilers. Their details are wholly technical, and I see no reason why an officer who does not have to study them should need any more than the fundamental principle upon which all are constructed. A set of tubes, in one form or another, which are kept full of water by a feed pump, is united to a cylindrical drum, where the steam (formed by the fire playing around the tubes) may have time to separate from entrained water. In some cases, water is also carried in the drum, and additional drums are fitted at the lower ends of the tubes for a downward circulation or current, as in the Thornycroft boiler. These boilers as a rule require greater skill and higher training for their successful operation. In service the tubes become foul with soot on the fire side and

with cylinder oil or other impurities on the water side. I have seen several types fail in emergency by the giving out of the tubes. They are very good boilers for torpedo boats and steam launches, where the runs are all short and where sufficient intervals of rest are allowed for cleaning.

The following different types in common use are merely mentioned, with the principal advantages and disadvantages, referring to the reports of the Engineering Bureau and technical papers for detail:

The locomotive boiler, such as we see every day on shore, is not well suited to the Navy. While forced draft produced by exhaust steam in the railroad service seems to work well, the systems of blower draft necessary, with silent exhaust and saving of water, have been the cause of many disasters in foreign torpedo boats. I do not know what has been the result of experience with these boilers on the *Lepanto* as a reserve. Mr. Yarrow has brought the locomotive boilers to their highest state of perfection, and is said to have constructed a torpedo-boat boiler of this type, entirely satisfactory for all speeds.

The Herreshoff boiler is the best known in the Navy. With good water, intelligent care and running, and regular overhauling, it is fairly successful with a life of two to four years. The coils are liable to corrode through near the joints and are troublesome to repair. Any carelessness is sure to cause burning of metal.

The Ward launch boiler is now replacing the Herreshoff, and although higher, is on the whole more satisfactory. It requires less skill and the danger from low water is not so great. The tubes, especially the hanging tubes which sometimes fill with thick oils and burn out, are easy to replace. The four boilers on the *Monterey*, which are of a somewhat different type, have been tested under steam, but have not yet had a full-power trial. They will probably do well.

The Thornycroft boiler is much used in England with fair success in torpedo boats. It probably requires less care than the Herreshoff, and leaky tubes are more quickly repaired.

The Belleville boiler is the only tubulous boiler used at sea

in large steamers. It is said that sea water can be used in it, but little information is given out with regard to its actual performance. The tubes are easily accessible, and the weight of the whole boiler and water is rather high compared with the other tubulous boilers.

The Towne boiler answers very well for small boats. Not a great many have been built, and I cannot say what they would do in a torpedo boat. The boiler is low and screw plugs on the outside give ready access to the tubes.

Of all the above boilers, I think the Ward boiler most satisfactory, on the whole, for steam launches. It is liable to foam with much forcing, but is reliable under all ordinary conditions of steaming. The weight is small.

AUXILIARY BOILERS.

I will add only a few words about the auxiliary boilers placed on board our ships. They frequently steam for long periods with very irregular duty during the day, and must be humored. Certain hours should be set apart for cleaning fires and tubes, getting out ashes and otherwise caring for the boiler, when no boats should be hoisted and no machinery run. This rule ought to be as inviolable as the men's meal hour. This boiler needs the same care internally as the main boilers, and is liable to the same disarrangements.

Not much has been said in the preceding on the subject of the design of boilers for forced draft, because it is purely a technical question which engineers have not yet succeeded in solving. Very moderate forcing, with $\frac{1}{2}$-inch to $\frac{3}{4}$-inch pressure in the fire rooms, can even now be used, and I look forward confidently to the time when 3 inches and 4 inches, or its equivalent, can be carried in the ash pits without damage to internal parts. Greater flexibility for adjustment to heat effects, combined with protection to the parts immediately in contact with the flame, will undoubtedly accomplish much. The English have saved weight of boilers by leaving out all division plates, required to make separate combustion chambers, and therein consists the main difference between their boilers and ours. As said before, the

matter of circulation among the tubes is highly important, particularly near the fire ends, and here we have been cautious not to overcrowd the tube box.

The use of a pump for circulation, while raising steam, which has been introduced in some of the late ships, has many advantages over the hydrokineter, although it is not of much use in raising steam before actually starting fires, as can readily be done with the hydrokineters.

LECTURE THIRD.

The Marine Engine.

The growth of the machine which has given us such triumphs of speed and safety on the sea, forms so fascinating a page in the history of mechanics that a temptation to dwell upon it is very great; but, before taking up the practical side of the question, I will content myself with only a few facts which have enabled us to go from 12 knots to 20 knots in the last fifteen years.

The contrast between the old and the new is very striking when we stop to consider it. The old fashioned engine jogged along comfortably at thirty or forty revolutions with twenty-five pounds of steam in the boilers, occasionally needing a push to get over the center. We have one of them left for use in the education of Naval apprentices here in the Bay. Now, the engines are models of beauty and speed. They go, apparently, without effort at any number of revolutions up to 100, employing steam pressures above 150 pounds. It was but natural that the change should come mainly from the merchant service where competition rendered great speed a necessity, regardless of cost of running. We have followed their lead, and commerce destroyers of small size have been built in the vain hope of exceeding the speed of the mail steamers by putting into light hulls many tons of machinery. The great outcry against the Advisory Board ships arose from the impossibility of using them to catch the fast Atlantic liners. (No ship could have been built of their size with sustained speed high enough.) Others were built to do it, and we really did succeed in passing the *Chicago* and others, but the merchant ship still remained far ahead. Now, the *Columbia* is approaching completion, and we hope to equal the sustained speed of the mail steamers. The later growth of ships is one of constantly increasing horse powers with hulls built to carry coal and the machinery. Twenty thousand horse power is required and the displacement has to be enormously increased to carry the engines and boilers. We are only now beginning to realize that we have to pay for high speeds in great weight of machinery

and enormous coal consumption. Let us see how some of the problems in engine construction have been solved by the engineer and how much reduction there has actually been in space occupied and weight for a given horse power.

There is a remarkable exception to our unsuccessful chase after merchant-ship speed in the case of the *Wampanoag*, condemned in 1869 by a board of officers on steam machinery afloat. Their report presents some very interesting points in the light of later developments of the Navy. The *Wampanoag* was condemned because, as stated in this report, it was "utterly impracticable to render her a vessel of war worthy of our Navy." She was calmly set aside "as a source from which important lessons can be drawn." I know of nothing more useful than the lesson to be learned to-day from a comparison between this ship and the *Charleston*, of about her tonnage. The former was begun just before the close of the war, with the avowed intention of building an armed ship to catch blockade runners and sweep the seas. Her success was in every way creditable to Mr. Isherwood, who designed machinery unpecedented in size and power. She steamed 16.71 knots for 37 hours, a performance that would do credit to the best ship of our Navy to-day, or to any man-of-war afloat, and at that time far in excess of any merchantman. I cannot help admiring the courage of the man who designed this machinery at a time when we had no iron hulls, when materials were poorly suited to the purposes, and machinery for construction was almost inadequate. The result achieved is enough to make any engineer's fame, although the ship as a whole was predestined to failure as a man-of-war. The only change we have succeeded in making since 1865 is in reduction of weights, as will be seen.

The *Wampanoag* had a displacement of 4,370 tons, with draught of 19 feet, 2 inches; length, 335 feet; breadth, 45 feet, 2 inches. The *Charleston* has a load displacement of 4,165 tons, draught 20 feet, 1 inch; length, 321 feet; breadth, 46 feet, 2 inches; so that they are practically the same. Now let us see how the high speed has been obtained in each case. In order not to rack the hull of the *Wampanoag* with high speed reciprocating

parts of the engine, gearing was adopted, as it was absolutely necessary to have a high-speed screw. The main engines made only 31 revolutions a minute, and the screw 63.7, on the trial trip. The boilers carried a pressure of 32 pounds per square inch. Cylinders 100 inches in diameter were provided to obtain the power required, and a stroke of 4 feet was given the piston. Compound engines were hardly known, and cylindrical boilers were just coming into use. There was no improved steel, and all parts had to be made very heavy, and the boilers built up of wrought-iron plates cut to small size. Notwithstanding these drawbacks, the weight of machinery and boilers was only 1,260 tons, and the weight of coal brought the total for propulsion by steam up to 2,010 tons, or .46 of the displacement.

The *Charleston* has compound engines, with low pressure cylinder of 85 inches diameter, and piston stroke of 3 feet, and large cylindrical boilers, carrying 90 pounds pressure. The total weight of machinery and boilers is 732 tons, which the coal will bring up to 1,492 tons for propulsion, or .36 of the displacement. The *Wampanoag* was actually condemned for excessive weight of machinery and coal, and some other deficiencies, possessed equally by the *Charleston*, whose sea speed is probably not so great. The exact gain in 20 years is represented by a saving in weight of machinery equal to $\frac{1}{10}$ the displacement.

To the improvement in the manufacture of steel and other metals, and the accuracy of our machine tools, may be attributed the greater part of our recent growth. Engineers have long realized that the true method of increasing power was by greater piston speeds. It makes no difference what the type of the engine may be, the horse power is directly proportional to the piston speed, if the same pressure or load can be kept on the piston. Of course, the limit was soon reached with simple engines and old style boilers by the excessive weight of boilers required to furnish steam to a wasteful engine. With the introduction of compound engines came greater economy, higher pressures, and therefore increased piston speed. As boiler work and steel have improved, higher pressures and longer grades of expansion have come into use, and triple and quadruple expansion engines have

taken the place of the older types. Pressures have run up to 180 pounds, and will probably go higher in time to come. The increase of boiler pressure does not signify that the piston receives a greater load, inasmuch as high grades of expansion mean low mean pressure. In fact, I think the crank-shaft moments have decreased and become more uniform with the increase of speed, thus rendering our engines even safer than those of ten years ago. When the limit of speed came in excessive weight of boilers, the blowers were again introduced to provide more steam behind the piston. (A reversion to an old type used during the whole of our civil war. An old officer has told me that they thought nothing of shutting up the ash pans and turning on the blowers full speed when they were reaching out for a blockade runner loaded with cotton.) There has been a constant decrease in the diameter and pitch of propellers for a given speed; also, as the limit of boiler construction has been reached from time to time, an increase in the size of ship to carry the machinery. This has been mainly the history of the merchant service, with keen competition, and we have followed; but the Navy would go through a wilder race for speeds, if we had three or four years of blockade running to prevent. One great advantage possessed by the engineer to-day is the solid foundation on which to place his engine and the rigid lines of shafting. Notwithstanding this, great care and judgment are needed to prevent disaster, demonstrated by the accident to the *City of Paris*.

I come now to the means by which we have achieved the gratifying results in our own ships. In the design of the engines, great attention has been paid to proportioning the parts so that the least weight can be made to do the work. The factor of safety has been reduced as the metals have reached greater uniformity and strength, and materials of higher quality have replaced cast iron wherever possible.

MAIN ENGINES.

The main cylinders have been increased in weight rather than reduced for a given steam pressure, since their proportions depend more upon the ability of the foundryman to obtain sound

castings than upon the actual strength required. Liners of metal specially mixed for good wearing surfaces are usually put in to form a steam jacket. All the moving parts are made lighter both by use of stronger metals and a reduction in the factor of safety. Bed plates are also reduced in dimensions, or left out altogether by reason of bolting the pillow blocks directly to heavy keelsons and flooring of the ship and tying them rigidly to the cylinders. The use of hollow shafting has made a slight saving of weight, but has subserved the far better purpose of laying the interior bare for inspection. The *Dolphin* would not have broken a shaft, if the center had been bored for examination, as the interior cracks would have been discovered, also the variation in structure of the metal.

The adoption of piston valves saves weight, but the change was not made entirely for that purpose. It was for the sake of complete balance from the pressure of steam, and therefore great reduction of friction. The piston valve has its faults in large clearances and considerable leakage of steam after long runs, which the ordinary slide valve would be free from. The packing rings require such frequent adjustment that solid pistons fitted accurately in the bore of the steam chest have been adopted in some cases lately. Radial valve gears have come into use and a few of our ships have the Marshall gear. These gears get their name from the radius link which provides the proper steam opening. One eccentric, or none, is used for the lap and lead. The steam distribution is not better than that of a well designed link motion, but there is considerable saving in length of engine as the valves may be put on the side.

Probably an old-time engineer would be struck more by the greater accuracy of all fitting and the better proportions of all journals to the work required than by anything else, also by the greater rigidity of the engine as a whole. No doubt there has been a reduction of weight in the machine itself for a given stress upon it, but the great saving has resulted from high piston speeds.

When we come to the machinery outside of the engine itself we find a material change. Instead of an enormous cast-iron

condenser supporting one side of the engines, we now have a brass cylinder, made as light as the requirements of service will permit, with separate air and circulating pumps. The advantage of the latter can hardly be overestimated. A good vacuum can be maintained at all times; the main engines have nothing to run but the screw, and, in action or emergency, all surplus steam can be turned directly into the condenser without hiding the ship in a fog bank. The change in the shape of the propellers has resulted entirely from the introduction of twin screws and high speeds. As most officers know who have seen a modern ship in dock, there are three blades on each screw, which have the general outline of an oval. The greatest breadth of blade is placed at the most effective part, in order not to expend an enormous amount of power in the friction of a broad tip, or in the side resistance of a long base. The hubs are also made much larger for the latter reason. The most striking feature, however, is the smallness of diameter, which has resulted in a great increase of efficiency. Such a thing as negative slip is no longer heard of, and we approach much nearer Froude's maximum efficiency with about 13 per cent. slip. The design of propellers as to proportion of pitch, diameter, and area for a given ship is not well understood yet, in spite of all the experiments carried on in England and this country, and many of our ships have had to experiment on the preliminary trials. The *Charleston's* blades had to be given a finer pitch and the *Baltimore's* a coarser, in order to obtain the required horse power. With every departure in the size and shape of hull, new elements of uncertainty arise which can only be decided by actual trial. For this reason, we may find it necessary to make more than one trial of the *Columbia* with her triple screws. There has been a great reduction of area as well as diameter. Mr. Seaton gives a formula for the proper area of blades,

$$A = K\sqrt{\frac{\text{I.H.P.}}{\text{Revolution}}};$$

and the value of K he puts at 13. As a matter of fact, this constant has gotten down to 8 for the *Concord*, 8.66 for the *Philadelphia*, while it was 17 for the *Trenton*.

The screws cast from high grade metals can now be made smooth, and the edges filed sharp. I do not understand the object of painting blades, as required by the Department, when there is such obvious advantage in polished surfaces, which collect very little scale and barnacle. The fear of galvanic action on the stern of the ship is probably at the bottom of it. I cannot believe that the danger is great with twin-screw ships. The covering pieces of the stern tube, and some parts of the strut, may corrode, but not enough to do any damage. Where cast-iron screws are used it is generally the backs of the blades that go. I have seen dozens of blades pitted almost through in a line passing diagonally across the back and terminating near the tip. It struck me as the line where the eddy resulting from the oblique motion of the surface through the water is formed—just the spot where air would be carried down by blades projecting above water when the ship is light, or pitching in a heavy sea; and this air may account for some of the corrosion at the stern and on the blades.

TRIPLE-EXPANSION ENGINE.

No attempt will be made to describe the different parts of an engine, with whose names every graduate of the Naval Academy must be familiar. The following remarks with regard to the triple-expansion engine, the latest bidder for our approval, may not be out of place: The triple-expansion engine consists of three cylinders of different sizes, through which the steam is passed on its way from the boiler to the condenser. Steam is admitted to the H.P. piston, and does a certain amount of work during the stroke. It is then exhausted into a small receiver, from which it passes to the intermediate cylinder, in which it expands to a greater volume and does additional work, and, then, to the L.P. cylinder, where it expands again and makes a third addition to the work. The cylinders are so proportioned that each will do one-third of the total work of the engine. The steam which enters the H.P. cylinder at a high pressure must expand in volume to fill the L.P. cylinder, and, thus, we have a certain rate of expansion independent of any cut-off on the engine.

As a rule, however, the cut-off is taken at .7 the stroke, in order to obtain a good distribution of steam, proper lead, cushion and exhaust. From ten to twelve expansions are considered the most economical rate of running a triple-expansion engine, but this must depend somewhat upon the initial steam pressure. No separate cut-off valve is fitted, and the cutting off is effected entirely by means of the link, with a slight adjustment for each cylinder separately provided in the reversing arm. A live steam pipe is led directly to each receiver, for the purpose of warming the engine and blowing water out; also for starting. By means of live steam and the adjustment provided above, the powers in each cylinder may be nearly equalized at all rates of running. This is usually done for smoothness of working, although it does not necessarily conduce to economy.

In designing the engine for a given power, the L.P. cylinder is proportioned as if it had the entire horse power to develop, supposing the steam were cut off and all the expansion took place within one cylinder. The other cylinders are then proportioned, both by comparison with other engines already built and running, and by theoretical calculation, to give equal powers. A good rule is to make the area of the H.P. cylinder one-fifth of the low, and the intermediate cylinder a geometric mean between the two divided by 1.1. Supposing D_1 = L.P. cylinder diameter, D_2 the intermediate cylinder, and D_3 the H.P. cylinder; then, $D_2 = \frac{D_1}{1.57}$, $D_3 = \frac{D_1}{2.24}$. The cranks are always set at angles of 120° apart.

The engine requires more care only by reason of the additional parts; but the separate cut-offs and all their necessary valve gears are done away with. The handling of the engine is easier than with the old type, because it has only the screw to run, so that after all the auxiliaries are started, the engineer can give his entire attention to it and the boilers while getting under weigh.

As stated before, the triple-expansion engine is not economical at low powers. The cylindrical surfaces are large, and the amount of steam used small, so that the percentage lost is very

great. Suppose, for instance, that 5 per cent. of steam is lost by condensation in the cylinders at 3,000 I.H.P. This amounts to 150 I.H.P. Now, if the engine is run at a low speed—say at 300 I.H.P.—and the same amount of condensation takes place, (it is usually greater), the loss would be 50 per cent. I cannot state the exact relative efficiency of simple, compound and triple-expansion engines at low powers, as I know of no exhaustive experiments on the subject.

In the armored cruiser *New York*, two sets of engines have been coupled to each shaft, with the intention of throwing the forward engines out for ordinary cruising. One great fault with this plan will be found in the great amount of wear on one engine, while the other, to which it will be coupled for full power, has had no wear at all. The shaft alignment may give trouble.

The triple-screw cruiser *Columbia* was designed with the idea of uncoupling the wing propellers for low speeds, and thus avoiding the great waste from running all engines. It is estimated that the drag of the screws will be considerably less than the friction and condensation.

The low-pressure cylinders are sometimes designed to be thrown out, in order to work the engines compound.

The use of triple screws on the larger cruisers, and even the smaller vessels, has many distinct advantages, if it can be made a success. The division of power among three shafts greatly reduces the danger of break-down and the weight at the stern of the ship. The screws may be made of much smaller diameter, and higher revolutions thus obtained on the main engines, which we have seen tends to economy. The uncoupling of two engines, when running at economical speed, will undoubtedly reduce the coal expense, although the slip of one screw and drag of the other two may be greater than we anticipate. Note that the screws must be placed well out of one another's way. In the *Columbia*, their projected circles do not touch.

It has been only recently that the merchant service has adopted twin screws, (in their case, copying a practice long followed in the Navy,) and now the *Gigantic* is to be built with three screws and 45,000 horse power for the Atlantic service.

It may be useful to know the metals of which the different parts of the engine are made, and I will give a brief list of them here.

Forged Steel.—Forged steel has largely taken the place of wrought iron. It can now be made of uniform quality in large masses, of strength far above iron. Piston rods, connecting rods, tie rods, shafts, and the numerous small forgings about an engine are all made of it; also, in some cases, the crosshead and framing. The reduction in weight of reciprocating parts resulting permits a far higher piston speed.

Cast Iron.—The use of cast iron for cylinders and wearing surfaces in all cases of sliding friction has not changed. It stands the heat well, and, having a low co-efficient of expansion, bids fair to remain the best material for all complicated castings liable to be heated to the temperature of steam, and for those parts which require rigidity rather than mere tensile strength. The trial trip of the *Charleston* should serve as a warning, however, against the use of steel and cast iron together for sliding friction.

Cast Steel.—This metal has a less extensive field of usefulness than was once anticipated for it. Many parts are successfully cast from it with a considerable saving of weight, but large castings usually have to be accepted with surface cracks and small blow holes in them. The test pieces often gave fine results, while the larger masses exhibited such imperfections as to render them unfit for use. Pistons, in some cases crossheads and engine framing, and parts of the valve gearing, are made of it; but the attempts to obtain castings of complicated shapes have generally failed, especially where there are great changes of shape or dimensions. One of the worst elements connected with this metal is the uncertainty of obtaining what you want. I have known builders to make certain parts of wrought iron or forged steel rather than submit to the many delays caused by the cast-steel men. As a wearing metal, it is practically worthless, on account of irregularity of structure and particles of grit often found within it.

Copper and Zinc Alloys.—The use of brass and composition is probably somewhat more extensive than in former years, but the

change has been rather in the adoption of different grades of material for different purposes. All brass bolts are usually of forged metal, which can be worked either hot or cold. Tobin's metal, (which contains about 2 per cent. of tin), and Naval Brass, (about 1 per cent. of tin), are in common use for this purpose. Propellers are now made of high-grade material, such as Delta metal or manganese bronze, which reaches fully 40,000 pounds tensile strength in large castings. The bronze formerly used for this purpose gave about 23,000 pounds. Rolled brass is now extensively used for condensers, which are made cylindrical. It has great strength and considerable stiffness, allowing the condenser to be made much lighter. Joints are usually brazed. The old navy composition is still used for bearings and all journals, but a lining of white metal to form the bearing surface is always put in. This metal is raised in blocks well above the surrounding brass, leaving large oil grooves. For this reason cast iron might as well be used for boxes, except for the difficulty of overhauling in case of corrosion in place. The introduction of white metal has been followed by the cessation of hot pins and bearings. Of course the question of proportion enters here also; but it is nevertheless true that we seldom hear of a hot pin now at any speed from 30 to 100 revolutions.

Copper.—This metal is used for all pipes, especially where many bends have to be made. It is very ductile, and can be worked readily into a great variety of shapes. The smaller pipes are usually drawn from the solid metal, and, therefore, have no joints; but the larger pipes, with longitudinal brazed joints, form a great element of danger on board our ships. Many fatal explosions have been caused both in the merchant service and the Navy, of recent years, by the giving way of large steam pipes, which have generally torn apart in the brazing. I might say here that no welded or brazed joint should ever be trusted for a tensile stress. The Bureau of Steam Engineering, warned by several recent disasters, has required all large steam pipes to be banded at intervals of six inches with steel strips two inches broad.

WEIGHT OF MACHINERY.

The following table of comparison between the *Trenton* and *San Francisco* is added for the sake of showing where the weights have gone and what we are doing in the latest ships. The greatest saving lies in the engine rooms, in spite of doubling the number of engines and tripling the number of cylinders. I have made out the weight per horse power both for maximum and sustained speeds, calling attention to the fact that the *Trenton* had no blowers, and, consequently, no maximum power comparable with that of the *San Francisco.* For the sustained speed, however, the weight of all machinery in the latter, per I.H.P., is less than half that in the former. It strikes me that a fairer comparison is based on the moment on the shaft, thus eliminating speed of piston, which forms the main element of difference in horse power. We thus see that at the highest sustained speed the weight of machinery is not much greater in the *San Francisco* for one foot ton of moment on the shaft; in other words, the *San Francisco's* machinery is as safe and reliable as the *Trenton's*, notwithstanding the great increase of piston velocity. But for maximum power the weights are only a little over one-half, and the engines are strained to the utmost within safe limits. Let it be noted here that the care and judgment required on the *San Francisco* is far greater, notwithstanding the greater safety at ordinary speeds, on account of the high steam pressure and the rapidity of motion.

	Trenton.	*San Francisco.*
Weight of boilers and appurtenances	476.8 tons.	465.0 tons.
Weight of engines and appurtenances	468.7 tons.	416.0 tons.
Weight of pumps and stores	39.9 tons.	33.1 tons.
Weight of all machinery	985.4 tons.	914.1 tons.
Maximum indicated horse power	...	9913.0
Sustained indicated horse power	2460.0	4957.0
Displacement	3814.0 tons.	4088.0 tons.
Revolutions per minute	53.77	124.8
Stroke	4 ft.	3 ft.
Foot tons of moment on the shaft, maximum	...	186.26
Foot tons of moment on the shaft, sustained	107.3	116.0
Weight of boilers, &c., per maximum I.H.P.	...	105.0 pounds.
Weight of engines, &c., per maximum I.H.P.	...	94.0 pounds.

	Trenton.	*San Francisco.*
Weight of all machinery maximum I.H.P.	...	206.5 pounds.
Weight of boilers per sustained I.H.P.	434.2 pounds.	210.1 pounds.
Weight of engines per sustained I.H.P.	426.8 pounds.	188.0 pounds.
Weight of all machinery sustained I.H.P.	897.3 pounds.	413.1 pounds.
Weight of boilers per ton of moment on shaft, maximum	...	2.497 tons.
Weight of boilers per ton of moment on shaft, sustained	4.445 tons.	4.010 tons.
Weight of engines per ton of moment on shaft, maximum	...	2.234 tons.
Weight of engines per ton of moment on shaft, sustained	4.369 tons.	3.395 tons.
Weight all machinery per ton of moment on shaft, maximum	...	4.908 tons.
Weight all machinery per ton of moment on shaft, sustained	9.186 tons.	7.890 tons.

The derangements occur more suddenly and may be far more serious in their results; add to this the care of a great number of auxiliary engines in operation.

VERTICAL ENGINES.

The most serious obstacle in the path of Naval Engineers has been the horizontal engine, a necessity with the old ships and many of the new in order to keep the machinery below the water line. The old type of return connecting rods with long side rods has departed with single screws, and direct acting engines are universally used for twin screws, situated on the opposite sides of the ship. A complete change has been made by the present Chief of Bureau, and vertical engines are to go into all ships large enough to take them, a departure which will undoubtedly give us simpler and far more efficient machinery. In some cases, the stroke has been made quite short, to get the cylinders below the protective deck. Where armor is used, there is no difficulty whatever in adapting the vertical engine to vessels of war.

The following advantages possessed by the vertical engine over the horizontal are given to show practically the advantage possessed by the merchant service over the Navy for many years, and what we may expect to gain.

1. The great decrease of friction of the moving parts. This varies greatly in different horizontal engines, but depends mainly on tightness of fit for vertical engines, where the weight is taken on the crank pin, and the surfaces can be so well oiled that the work of friction is exceedingly small. On the contrary, with a horizontal engine the weight of all moving parts is taken on cast-iron surfaces, where the friction is entirely a sliding one and the oiling arrangements often of necessity ineffective. The weight of all reciprocating parts of the *Charleston* is 18,584 pounds, which absorbs 20 I.H.P. for the friction due to supporting the parts on horizontal surfaces alone, at the most economical speed, supposing everything well oiled. This represents about half a ton of coal per day.

2. The tendency of cylinders and piston-valve chests to wear true and straight. In a horizontal engine the bores all wear down and out of true, necessitating frequent relining.

3. The tightness of all steam pistons. Where the cylindrical surfaces wear out of true the packing rings stand off at the top, and allow more or less steam to leak by into the exhaust passages. This becomes a serious matter after long runs, and decreases efficiency enormously. Another point, probably as important, is the great increase of friction resulting from lining up and setting out the packing rings to fill an oval, in order to check the leak. A shoe is usually put under the piston, and also an extension piece, which prevent the use of the best forms of packing rings.

4. The packing rings can be made practically independent of the piston, and, therefore, a very uniform pressure on the cylinder surfaces obtained. Any slight change in alignment does not affect the tightness, and the piston and rod may actually be thrown out of the center by the stuffing box without any increase of wear on the ring.

5. The greater durability of packing, inasmuch as there is no tendency for the rods to squeeze one side of the stuffing box more than the other. In a horizontal engine the rods are sure to go down and the oil is equally certain to seek the lowest part of the packing.

6. The use of the cylinder oil is greatly reduced, or done away with entirely. I think this is probably due to a slight water packing resulting from condensation on the inner surfaces. This water packing is nothing but a thin film of liquid which settles down around the packing ring and probably helps to keep it steam tight.

7. The pressure of steam and weight of reciprocating parts is all taken in the crowns or bottoms of brasses, allowing much more satisfactory adjustment. With horizontal engines, the brasses tend to wear in the sides or joints, and frequent refitting becomes necessary.

8. A better design of valve gear and engine framing is possible, inasmuch as all pressures are taken vertically.

9. The jar due to the rapid motion of reciprocating parts is all up and down in the line of greatest strength of the ship.

10. The water can be gotten out of the cylinders with greater ease, as the lower exhaust port forms a natural drain, and very little collects above the piston.

11. Less space taken in the ship, since the engines can be placed abreast of each other, instead of one foward of the other.

12. The great accessibility of all parts in overhauling and lifting. Everything is a straight lift for which permanent provision can be made. The difficulty of removing a cylinder head or piston from a large horizontal engine can be appreciated only by those who have had it to do.

CARE OF MACHINERY.

I cannot add anything to the directions given in the Engineer's log book for the care of machinery. Examination of the internal parts of the engine, careful adjustment of all wearing parts, the quarterly examination of holding-down bolts, pillow blocks and all standing parts; these, with thorough cleanliness and plenty of paint, will make any engine as efficient as man can make it. The bleeder valves should never be used except in case of necessity, as the hot steam may cause the condenser tubes to crawl and get out of the tube sheets. The reversing of main engines should never be sudden, especially at high speeds.

There are cases, no doubt, where an emergency justifies it, but under ordinary circumstances of cruising, coming to anchor and mooring, there is generally plenty of time, and the engines should be spared as much as possible.

So long as the above rules are followed, the main engines will take care of themselves, of course, supposing that proper judgment has been exercised in the overhauling. The great source of anxiety and trouble for an engineer on board ship lies with the auxiliaries, pipes and attachments. It is always some little thing that gives out in emergency. A serious breakdown, or great detention, of any of our new ships on account of the larger machines is reduced to a minimum by having twin screws and all important parts duplicated.

AUXILIARIES.

The old auxiliaries are well known, but I will name them in connection with others more recently placed on board ship, with derangements they are likely to incur.

CIRCULATING PUMP.

This pump is nothing more than a rotary fan working in sea water, and driven by a plain direct-acting engine, either simple or compound. The exhaust may be led into the atmosphere, condenser, or receiver. Its advantages are evident in the absence of shock from reciprocating parts and the total immunity from rupture of the condenser bonnets if the outboard delivery is left closed. In the latter case, the fan simply churns around in the water. The engine is always simple in all its parts and easy to run. There are no valves and pistons to overhaul. A connection for taking water from the bilge or any compartment is always fitted.

AIR PUMP.

The separate air pump, comparatively recent, is a great convenience in making a vacuum and clearing the condenser before the main engines are started, also for maintaining the vacuum, if at any time they are stopped. The work required of it is very irregular at different parts of the stroke, which makes close watch-

ing a necessity when steaming, as some of these pumps have an ugly habit of stopping. Its adoption followed naturally the high-speed engine, since high speed in a reciprocating pump connected with the main engine is equivalent to inefficiency, where the difference of pressures in the suction side and the condenser are so small. The vertical pump is preferred for the water packing on the pistons and the more direct discharge of water and vapor. The exhaust is usually led direct to the condenser. The valve, stud bolts and guards have to be looked after very carefully, and any sudden leakage from the tubes is sure to overload the pump. Several narrow escapes from complete breakdown have occurred on trial trips from the latter cause.

MAIN BILGE PUMPS.

In many cases, this pump is connected directly to the air pump, but a separate pump, worked from its own steam cylinder, seems preferable. It is generally of the plunger type. Strainers to intercept coal dust, ashes, and bits of waste or wood, are placed in the suction pipe, notwithstanding which the valves wear rapidly, and frequent overhauling is required. Constant watchfulness is necessary to keep the bilge free of water under weigh, as the smallest particles under the suction or delivery valves destroy the efficiency. Connection is made with every bilge compartment.

BILGE AND FIRE PUMPS.

One of these pumps is usually placed in each engine room. Its connections are the same as the main bilge pump, with the addition of a sea suction and delivery into the fire mains; also, into the auxiliary feed pipe in some cases, and the same provisions against choking pipes and valves with dirt have to be made. These pumps are run every morning for washing decks, and often for other purposes. A spring safety valve is placed in the delivery pipe as a relief in case of overloading. Where the pump is fast running, the outlet valves on deck should not be closed before stopping the pump, as a joint may blow out or a bolt break before the relief valve has time to lift. I have spent

an entire day repairing damages on the *Charleston* caused by a piece of carelessness of this kind.

FEED PUMPS.

In general, there are two feed pumps in each boiler compartment, one main and one auxiliary, of the vertical, duplex type. They are usually fitted with metallic valves, to withstand the heat, and, since there are from four to eight of them in every ship, anyone of which may feed any boiler, the immunity from derangement is reduced to a minimum.

The main feed pumps take their suction from the feed tanks, and deliver into the main feed pipes only, being therefore used exclusively for feed. The auxiliary pumps are general utility pumps, having suctions from the feed tanks, bilge, sea, and boilers, and deliveries to the deck, sea, and boilers. After pumping the water from the bilge, the suction should always be put on the sea before feeding boilers.

FLUSHING PUMP.

The distiller circulation pump is arranged to deliver into the service pipe for flushing water closets, either through or by the distillers. It is constantly in use, and some other pump is usually connected as a relief in case it gives out.

PUMPS AND PUMPING SYSTEM IN GENERAL.

The type of pump is usually not specified in the above cases, and we have several varieties in the service. Some form of vertical pump is generally adopted, of the duplex type, for the small amount of floor space occupied by it. It may be well to state the two main classes into which all pumps divide themselves, (1) those with fly wheels, shafting and eccentrics, (2) those with only reciprocating parts and some form of tappet for working the steam valves. The former are said to be positive in their motion, and, although they may become noisy after a little wear, they rarely stop or refuse to work. The latter frequently give trouble in the smaller parts of the valve and gearing after a few years' service, yet, if duplex, I think they are superior for our purposes, being usually noiseless and easily taken care of. All parts in contact with sea and bilge water must be made of

bronze, and any small bolts or rods of steel or iron exposed in the pump chamber are soon eaten away. This is the nearest approach to galvanic action I know of on board ship, and yet, when it is remembered that bilge pumps are sucking air half the time, or a mixture of air and water, the cause of corrosion will be self-evident. They require regular overhauling and careful watching for corrosion.

The pumping system for compartments, bilges and double bottoms requires considerable attention, and should be thoroughly understood by the commanding officer. The systems are much alike in all the new ships, and I will describe that in *No. 13* only as an example. Provision is made (usually by sluice valves) for draining water from the store rooms, chain lockers, coal bunkers and other compartments throughout the ship, to places whence the water can be pumped either by hand or steam pumps. Each principal compartment within the limits of the double bottom, as well as those before and abaft these limits, has its own means of pumping provided. Each compartment of the double bottom is furnished with its own suction, led to a valve box in one of the engine rooms, whence the water can be drawn by either hand or steam pumps, and strainers are placed over the pipe connection in each compartment. The valve boxes have sea connections, so that the double bottoms can be filled by the channels used for emptying them. All these pipes are led above the inner skin where possible.

Two pipes, one 14 inches and one 7½ inches diameter, are led above the double bottom throughout the machinery compartments, with separate valves, to a well in each compartment. They also connect to a valve box in the forward fire room, and one in the after engine room, for draining the compartments in the extremities. The larger of these two pipes is connected with the circulating pumps of each engine room, so that any compartment can be drained by these pumps, which can throw an amount of water equal to the ship's displacement in one hour and a half or less. Connections are also made with all other pumps except the main feed and air pumps. Self-closing valves are fitted in the suction of each compartment, so that the water

cannot work back, and these valves are provided with means of closing from the gun deck. The smaller pipe is for general use in pumping the bilges. It has a suction led to each compartment, and the valve boxes above mentioned; also connections with all main and auxiliary bilge pumps. There is also a separate and independent suction direct from the bilge to the auxiliary pump located in its own compartment. The pipes in the general drainage system are of galvanized iron, with suction lengths of cast iron extending into the bilges. The hand pumps located on deck have long suctions to the drainage system. They rarely work well. I have seen several barrels of water poured down one of these pumps in the vain attempt to make it work.

The wells, as provided in the fire rooms, often choke up with ashes and dirt, and I would strongly advocate additional means of pumping, whose nature can best be shown by an accident which occurred on the *Charleston* during one of her runs from San Francisco to Honolulu. We had been five days at sea, and the bilge suction pipes in the after fire room were obstructed by soot and ashes. There were about three inches of water in the bilge, and a small block left under the boilers, washing back and forth across the ship, struck a drain cock and opened it. We had to haul fires on the boiler, as the bilge filled so quickly with hot water that nobody could get down to locate the trouble. The rush of water to the well filled it with dirt, and we then had no means of draining the bilges. It took twenty-four hours to get the water out and the boiler ready for use again. This suggests the fitting of an ordinary suction hose, which can be put down in any part of the bilge, to one pump in every fire room compartment for just such emergencies. The Union Iron Works voluntarily made provision for such a fitting on the *San Francisco*, I believe.

Now, a word in regard to taking care of the double bottoms. Considerable delay results from placing the manhole doors through the inner skin too near the center of the bilge, where a few inches of water may prevent a passage. All these doors should be as far up in the wings as possible, or be raised six

inches above the skin, in order to make the double bottoms accessible even at sea. I speak from experience here, having found it impossible to keep the *Charleston's* double bottoms as they should have been kept. The ship steamed a great deal, and a few compartments were filled with salt water at the end of nearly every run. Four of these compartments could not be entered with water in the bilges, so that we could not work in them at sea, and we had no men to spare in port, where the stays were often short. The Department provides portable lights for use in the double bottoms. Where a good system of ventilation is used, they are entirely unnecessary, as candles answer every purpose, and the tallow can be swept out after cleaning. There is almost always some water in the bilge or double bottom, so that the wires get wet and the current short circuited. A fireman cannot short-circuit a candle, and he is entirely familiar with the use of it. As an example of a good ventilation system, I will mention that we fitted on the *Charleston* an ordinary eighteen-inch centrifugal fan, belted from the forward circulating pump, which delivered air into the valve box leading to the double bottom. In this way air could be forced into any compartment through the filling pipe by simply opening the proper valve. A cock was provided for draining the valve box thoroughly before starting the fan. This was found to keep the double bottoms fresh, and it could be turned on at any time without opening a manhole plate. Its use with candles strikes me as much more practical than electric fans and lights carried around through the bilges.

MAIN BLOWERS.

The forced draft blowers are simply centrifugal fans driven by a simple or compound engine, and delivering directly into the fire rooms or into air ducts. Inasmuch as they are located in the fire rooms where more or less coal dust and ashes can reach them, they require very close attention when running, and careful adjustment. Enough has been said about the forced draft on board our ships, and I will only add that the blowers should never be run if natural draft can be made to do the work. It may be a good plan when using dirty coal to

run them ten or fifteen minutes every watch at a low air pressure to clean out the tubes; this saves labor in sweeping.

WINDLASSES.

There are fitted on deck from one to three boat winches, and one anchor windlass, which generally give no trouble. They should always be run by an oiler detailed for that purpose by the chief engineer. The cleaning and care of these machines can always be given to two or three men from the Engineer's force, who run them.

AUXILIARIES IN GENERAL.

The other auxiliaries, and the pipes and valves throughout the ship require constant attention from the engineer, and make a great part of his work on board ship. Where 160 pounds pressure is carried in the steam pipes, every joint and bolt will develop weaknesses in the course of time. All valves, especially the safety valves, need periodical grinding in and packing, and all the water valves located in the bilges have to be taken out about once a quarter. The whole auxiliary plant, consisting of boilers, pumps, evaporators, distillers, ventilating fans, blowers and heaters, require nearly as much attention as the main machinery. The evaporator coils have to be taken out often and scaled.

That we have every reason to be satisfied with the results obtained from our own designs, I think will be demonstrated by a perusal of the English papers. Hardly a week passes that we do not hear of some failure, or radical modification needed to make their new ships successful. Take, for instance, the following extract from "Industries" for August:

"It is also essential that perfect reliance should be placed in the component parts in an emergency, and that each individual ship when called upon should be able to fully and satisfactorily meet the requirements for which she was designed, particularly with regard to speed and endurance under steam. These special qualities can only be maintained so long as everything goes well with the machinery and boilers, but, unfortunately, breakdowns of these vital elements are not of unfrequent occurrence, and

vessels of the most modern types, engined by the most eminent firms in the country, fail at the critical moment when all the power that it was intended to develop is needed.

"For some of these failings the double-ended boilers have been held responsible, and justly so, but in other instances the desire to attain maximum results has induced designers to cut down the weights of the machinery, and this has been carried beyond the limits of prudence, in fact, to such an extent, that a point has been reached far below what is absolutely necessary for strength and safety. As a consequence, the engines in actual service have proved a constant source of anxiety to those entrusted with their charge. The defects developed in the first-class armored cruiser *Aurora* point definitely enough to the fallacy of economizing weight too rigidly in the engine room, and the utter uselessness of ships which only obtain the contract speed and power for a short time, instead of fulfilling under ordinary conditions of service the steaming capabilities they were originally intended to perform. The *Aurora* is practically a new vessel completed in 1889. * * *

"Ever since her completion, difficulties, more or less serious, have been experienced in the engine room, principally through defects which revealed themselves in the pistons and piston rods. Last summer new pistons were supplied. Previous to resuming her station with the Channel squadron, she was subjected to trial, and it was anticipated that no further defects of a serious character would be developed. Her commission, however, was destined to be a brief one, as in endeavoring to fulfill a full-power natural-draft trial in the Mediterranean in March last, a lamentable breakdown occurred through the bending of the piston rods; the trial had to be abandoned, and finally she was sent to England to undergo thorough repairs. New piston rods of increased diameter have been made and other important modifications are at present being effected in the valve gear, feed pumps, etc. With the stronger piston rods and the other alterations adopted, it is to be hoped that the machinery will be perfectly reliable in future. Her previous failures are indeed to be regretted. The blame seems to rest, however, with the design-

ers and not with the makers of the machinery. We hope that the experiment of reducing weight beyond the limits of safety in the *Aurora's* engines, costly as it has been, will not be without good effect in future designing."

Such cases as the above justify the conservatism of our Bureau of Steam Engineering with regard to cutting down weights. We have heard nothing of the so-called defects of machinery, but it is only justice to say that we owe a debt of gratitude to the English Navy for their mistake. We learned more from their breakdowns and failures than from their successes during a period when we had no ships. The above case only emphasizes the care necessary to keep our machinery up to its highest condition after it has been designed and built, particularly in the small things.

COMPARATIVE DIMENSIONS OF THE *WAMPANOAG*, *TRENTON* AND *CRUISER NO. 13*.

For the information of the younger men of the Navy, who have not access to the records, I add here a brief history of the *Wampanoag*, with a table of comparative dimensions. The *Wampanoag* was designed in 1863, the hull by Mr. Delano and the machinery by Mr. Isherwood. As Mr. Welles, the Secretary of the Navy, stated in his report, she was intended "to sweep the ocean, and chase and hunt down the vessels of an enemy." The hull was built at the New York Navy Yard, and the machinery at the Novelty Iron-Works, and she had her full power steam trial in 1868. None of the other competitive ships were successful. Her hull was of wood, and the principal dimensions are given in the table. The engines were direct acting, with the cylinders on one side of the ship and large gear wheels on the other. The screw was geared to make slightly more than two revolutions for every revolution of the engine. The boilers were of the old Martin or vertical water tubular type, designed to carry 32 pounds of steam, and the draft was entirely natural. Her trial lasted thirty-seven hours, and the result shows a speed far in excess of anything, either merchant ship or ship of war, afloat at that date. As a steamship, she was an unparalleled success, whatever may be said about the weakness of the wooden

hull and the vulnerability of the vital parts. She was the pioneer of the fast steamers, so far antedating them that she has been forgotten. It was a stroke of genius which put geared engines with slow reciprocating parts and rapid rotating screw into a ship that would have been racked to pieces by the engines we construct to-day. She was condemned in 1869 by a Board of Naval Officers for various reasons, among them the following: "Excessive weight of machinery and coal." *Cruiser No. 13* has 3,971 tons devoted to this purpose, about the displacement of the *Wampanoag*. "Enormous coal consumption." She burned 136 tons of coal a day, while our latest cruiser will burn 360 for her sustained speed. "Because her grate surface was 1.53 times the midship section." "Carried 2.6 months' provisions for the crew." (This was denied, and put at 6 months by a minority report.) "Relative length to breadth said to cause inordinate rolling and dangerous straining of ship." The later *Cruiser* has about the same relative dimensions and a lower coefficient of fineness. "Carried coal enough to last only five or six days at maximum power." The *Cruiser* has the same endurance. "Four-bladed screw nullified the use of the canvas." "The substitution of a two-blade screw recommended, and the removal of several boilers for the purpose of giving her full sail power." Her later sister carries no sails at all. Her speed was superior to that of most of our cruisers, and we have gained between two and three knots by building a ship of twice her size. Nobody thought of the fact that she had no water-tight subdivision, and would fill from one hole in her side or bottom.

No more striking contrast between the ideas of a good cruiser twenty-five years ago and what is considered a good cruiser to-day could be devised than this report on warships of 1869 when compared with the requirements for our new ships. The three ships given in the table are the most advanced types of their date and the *Wampanoag*, which was condemned then as worthless, was similar to *Cruiser No. 13*, in many particulars. There is no question of the superiority of the modern ship, but it springs entirely from the improved materials placed in the hands of the Naval Constructor and Engineer, and in no way

COMPARATIVE DIMENSIONS OF THE *WAMPANOAG*, *TRENTON* AND *CRUISER NO. 13.*

	1865. *Wampanoag.*	1875. *Trenton.*	1892. *Cruiser No. 13.*
Length on L.W.L.	335′	253′	412′
Greatest breadth	45′ 2″	48′	58′
Mean draught when loaded	18′ 6″	20′ 8″	26′
Greatest displacement, in tons	4370	3814	8600
Greatest immersed section, sq. ft.	740.5	735	1343
Length ÷ breadth	7.417	5.271	7.104
Coefficient of fineness	.566	.593	.487
Number of watertight bulkheads	None.	None.	15
Number of double-bottom compartments.	None.	None.	27
Sail power	Half.	Full.	None.
Complement of men	375	480	456
Length of ship occupied by machinery	166½′	96′	210′
Coal capacity, in tons	750	332	2000
Average coal per day, sust. speed, tons	136	60	360
Type of engines	Simple.	Compound.	Trip. Expan.
Number of cylinders	2	3	9
Diameter of cylinders, inches	100	58½, (2) 78	42, 59, 92
Stroke of piston	4′	4′	3½′
Condensing surface, sq. ft.	7168	8641	28422
Screw propellers, number	1	1	3
Screw propellers, diameter	19′	19′ 6″	14′—15′
Screw propellers, pitch	25′	25.35′	20′ 6″
Revolutions of screw per minute	63	53.8	117
Revolutions of cranks	31	53.8	117
Sustained sea speed, knots	16.76	12.61	19.50
I.H.P. sustained	4049	2460	15000
Boilers, number	12 (4 superheat.)	8	10 (2 aux.)
Furnaces, number	58	24	68
Grate surface, sq. ft.	1128	492	1522
Heating surface, sq. ft.	31148	12860	49250
Boiler pressure, lbs	32	80	160
Weight of all machinery, in tons	1260	985	1971
Weight of machinery ÷ displacement	.29	.26	.23
Weight of machinery and coal ÷ disp.	.46	.35	.46
$\frac{D^{\frac{2}{3}}S^3}{\text{I.H.P.}}$	303.4	199	199
$\frac{AS^3}{\text{I.H.P.}}$	861.	599	644
Battery	10 9″-smooth-bores, 3 60-pdrs. 2 24-pdrs. 2 12-pdrs.	11 8″-rifles. 2 20 pdrs. 2 12-pdrs. 1 3″-rifle. 1 3″-how'r. 1 gatling. ...	4 6″-rifles. 8 4″-r.-fire. 12 6-pdrs. 6 1-pdr. 2 ma. guns. 1 field gun. 6 tor. tubes.
Weight of Ordnance, in tons	111	288	203

Many of the data under *Cruiser No. 13* are based on estimates, as the ship has not yet been tried.

from higher mechanical ability. I do not wish to cast any slur upon the gallant officers who composed the board of condemnation, for I am sure they would be the first to point with pride to our achievements in '68, if assembled to examine a new ship just out of the ship yard. We have learned now that we must pay if we want a high-classed ship, and there is no grumbling over the coal account. Sails are no longer thought of for fast ships and iron-clads. Twin and triple screws may be thanked for the feeling of security under steam power alone. The water-tight subdivision with double bottoms has been introduced within the memory of the youngest officers. The great reduction in weight of machinery has made room for the crew and provisions, but the whole ship is practically given up to machinery. There will be, on *Cruiser No. 13*, about sixty-one separate engines, not counting cylinders, which would probably run up to 120, for main, auxiliary and pumping purposes.

As a curious piece of information, I add the following data on the machinery of *Cruiser No. 13*, in a form different from the ordinary tabulated statement.

Her low-pressure piston, which is 92 inches in diameter, has an area of 46 square feet, a very comfortable 6 feet by 8 feet state room on board ship, and this piston has an initial load of 100 tons, equal in weight to three locomotives. The mean piston speed at maximum power will be 11 miles an hour, and the maximum speed 16 miles an hour. The tip of the propeller blades will move through the water at the moderate rate of 75 miles an hour. The condenser tubes, if placed end to end, would form a tube 33 miles long, and, if flattened out, would cover about ⅔ of an acre. The cooling water passed through these tubes will be equal to 36,000,000 gallons per day, enough to supply a large city with water. The main boilers, if placed end to end, would form a tunnel 156 feet long, and large enough for a train of cars to pass through. If divided up into rooms, they would supply a hotel with 16 fair-sized bed rooms. The heating surface is equal to 1⅛ acres. The grate surface, if arranged on one grate, would equal a small town lot of 20 feet front and 77 feet depth. The boiler tubes, placed end to end,

would be 13½ miles long. The blowers are capable of supplying 84,000,000 cubic feet of air an hour, which would supply a good-sized yacht with a 10-knot breeze.

The coal required for a full-power run across the Atlantic would supply 150 families for one year in New York State. With 20,000 horse power she would lift herself (7,500 tons weight) to the Brooklyn Bridge in three minutes, if hoisting ropes were coiled around drums on the shafts. If the engines were set up on shore, and used as a catapult, they would throw a 300-pound weight with such velocity that it would go off into space entirely clear of the earth's influence. The fictions of Jules Verne sink into insignificance by the side of this coal-eating monster of the nineteenth century.

LECTURE FOURTH.

Miscellaneous Items.

COAL.

The kind of coal purchased is often determined by what can be found in a foreign or home port. Where there is a choice, however, good bituminous or semi-bituminous coal will be found generally most satisfactory for all the ships. Anthracite is rarely preferable, and never for ships with gunboat boilers, where it is likely to be exceedingly wasteful. The report made by a Board of Engineers on coals in 1885 should be read by all officers, as it contains reliable information on this subject. The conclusions in regard to coal for our warships are brief and probably correct, and I have therefore made the following summary of the relative merits of anthracite and bituminous coal taken in great part from their report.

Bituminous coal requires the least working, is easier on the men, and more economical. It forms a solid clinker which is easily removed without hauling out a quantity of half-burned coal.

Anthracite, on the other hand, is difficult to handle, and there is often much waste from cleaning. It also requires an appreciable time for ignition, usually from twenty to thirty minutes to reach a condition where combustion is uniform, along with a much higher temperature. This might be very important in a chase or where it becomes desirable to increase the boiler power very quickly. It is also more unmanageable and uncertain when coming into port, inasmuch as it lasts longer and makes the steam difficult to get rid of. With free burning coal a good flame and heat can be obtained within five minutes after throwing it on the grate.

Anthracite occupies as much space as semi-bituminous and less than some kinds of foreign coals. The difference does not amount to much however. In the *Columbia*, which carries two thousand tons of anthracite, we could stow eighteen hundred and thirty tons of the lightest coal ever bought for the Navy, a

difference of only one hundred and seventy tons, which would probably be made up by increased economy.

In the absence of smoke, which is thought to have great advantage for war purposes, anthracite is not superior to some kinds of Welsh coal, and very little superior to Cumberland. This advantage does not strike me as important enough to outweigh the other considerations of economy and quickness of ignition. The soot formed in the tubes from bituminous coal can readily be burned out by putting on the blowers for a few minutes, and thus the advantage possessed by anthracite in cleanliness disappears.

The two kinds of coal require about the same amount of air for complete combustion—roughly speaking, about 12 pounds for each pound; but they require this air to be supplied in different places. Anthracite, which has no volatile matters, must be supplied with air on the grate, where the combustion takes place slowly, at an intense heat; consequently, the design of the grate is most important, and the combustion chamber and tubes may be made relatively small. Bituminous coal contains a large amount of volatile matter, which is driven off at a low heat, and escapes up the chimney unless supplied with hot air after it leaves the grate. This coal may be heated in closed retort, and the gases escaping supplied with air only at the mouth of the furnace, where they burn with intense heat. Anyone who has ever seen a Siemens-Martin furnace is familiar with this. The grate is, therefore, unimportant, while a large combustion chamber, in which the volatile substances can mix thoroughly with air and burn completely, is the essential feature of the boiler. Officers who have been on the Pacific coast are familiar with black diamond coal, which burns at the top of the smoke pipe. There is no flame inside of the smoke pipe, however; it is only that the volatile gases escape too rapidly for the air supply, or the combustion chamber is too small for thorough combustion, and the hot gases therefore ignite when they strike the air. It is an actual fact that the blowers sometimes conduce to economy with such coal by producing more complete combustion within the boiler. We gained a knot in the *Charleston* on the same

coal expenditure per day by simply running the blowers slowly. The reason for this is apparent in the better supply and mixture of air.

Intense local heat of anthracite is also a great objection to it in the new boilers. The design of these boilers for forced draft is not favorable to the use of anthracite. The smoke pipe and calorimeter of tubes is generally too great. It will be found, therefore, that additional boiler power will be required for the same speed when using anthracite.

Another point that it may be well to call attention to is always to buy the best coal for long passages. A great saving of money, however, is often effected by buying cheaper coal where the distance is well within the steaming radius. For instance, the black diamond coal purchased at Seattle for the *Charleston* cost three dollars a ton, making the total expense for the passage to Honolulu seventeen hundred and sixty-four dollars for coal alone. The return passage to San Francisco cost, at ten dollars a ton, just fifty-four hundred and thirty dollars,

All coals must be free from sulphur, which rapidly attacks surfaces exposed to the flame. It must also be dry and free from iron pyrites, if spontaneous combustion be feared. The cause of spontaneous combustion is not clear, and I can only attribute it to some form of oxidation, aided by heat, moisture and iron pyrites. There is no doubt that combustion can be caused by starting this chemical action, which will then generate heat enough to hurry on the process until actual flame and smoke appear. I have seen a clear case of this on the *Hartford* and, lately, reports have come from the *Philadelphia* and *Charleston* of like cases. In purchasing coal, therefore, avoid all impurities; be careful to keep the coal dry, especially in hot weather, and use the coal out of the bunkers liable to be heated by proximity of boilers, as soon as possible.

Anthracite does not appear to deteriorate in the open air as rapidly as bituminous. This is not an important matter when coal is actually stowed in the bunkers, but it becomes a matter that should be examined when buying coal long out of the mines, especially if not under cover.

Many of our ships have coaling ports through the sides and permanent chutes, which undoubtedly is a more cleanly way of coaling. I do not think that as much coal can be stowed, however, as by the berth-deck scuttle, as the chutes take up space in the bunkers, and more room has to be left for the men to get out. Generally speaking, a ton's space will be left in each compartment for the latter purpose, so that the loss increases with the amount of subdivision. In the new ships the separate bunkers number from twenty-four upward. Besides the loss of space, the difficulty and tediousness of stowing so many pockets are very great. Toward the end, when only a few men can get into the bunkers, where they have to lie down, the work drags fearfully. It should be remembered that, in such cases, forty or fifty men are in spaces almost devoid of air and entirely filled with coal dust, stowing all the coal thrown in by two hundred on lighters. However, I have seen seven hundred tons stowed in thirty-six hours, when there was a press for time. The bunkers above the protective deck are usually arranged to afford as much protection as possible against shot, and below that deck for the greatest convenience to the fire rooms. The coal is carried in trolleys, or shoveled to scuttles by the coal heavers when steaming. The difficulty of getting coal out of the numerous compartments and to the fires has been noted in our Navy, and fully commented on in reports of the English Navy manœuvres. When steaming full power, men will have to be detailed from the deck to aid in striking down coal, and the Navy Department should authorize extra pay for that purpose. It is close, confining work, and the landsmen can hardly be expected to do coal heavers' work cheerfully without extra pay, even though it be a necessity. The coal should always be taken from the highest part of the protective deck first in order to increase stability, and be left in the lower angles of the wing bunkers to the last. There is also a great amount of extra work due to the constant demand to keep the ship on an even keel.

ENGINE ROOM SIGNALS.

The necessity of having absolutely reliable communication between the deck and the engine room will be at once recog-

nized by every officer. The engine room signals should be of the best possible type, and set up in such a way that there can be no mistaking the signal. A mechanical telegraph with which officers are familiar is most used in the Navy, and I think it is probably the most satisfactory, although the electric signal has been proposed. The mechanical telegraph is always there and ready for use, and any derangement is indicated at once by increased stiffness. A nervous man on the bridge can work off his energy on the handles without doing any harm, and as they are set to move in the direction he wishes the ship to move, mistakes can be only of degree. The only fault I have to find with them is that there are rarely enough of them placed on board ship. We had three on the *Charleston* all bunched together forward where one shot could have carried them away, and none on the after bridge. There should be at least three entirely independent telegraphs above the protective deck placed as far apart as possible, one on the forward bridge, one on the after bridge, and one near the engine room hatches.

There should also be speaking tubes direct to each engine room from every one of these places. These tubes can be made use of in the thousand and one cases where messengers are now employed. Stoppages and the various matters to be reported to the officer of the deck have been sent up by a coal heaver from time immemorial, but the tube forms a pleasanter and quicker messenger. It will also enable the officer of the deck to notify the engineer on watch, as long as possible beforehand, of any intended stop and the length of time the engines may be stopped, in order that he may provide for his fires and steam. All communication between the engine rooms and other compartments under the chief engineer's charge is by means of speaking tubes. Where the fire rooms are under air pressure these tubes are not of much use, as any one who has received a charge of coal dust in the ear can testify. Small bells or gongs with a set of signals arranged would probably do better.

Another set of signals which I feel ought to be spoken of here is for use in clearing the compartments when abandoning ship or closing water-tight doors. It is absolutely necessary to give

the men who work below confidence that they will be thought of in an emergency. On the *Charleston*, we had no signals in the coal bunkers or compartments, and the steam siren was used for warning the men. Frequently, it could not be heard at all, or only upon the second blast. After some of the men had been shut up in a wing passage alongside of the boiler, they did not volunteer to help close any more water-tight doors. I am convinced that a reliable gong, placed in every compartment, would be of far more use than the numerous water and fire annunciators placed in the coal bunkers.

SAFETY TO MEN.

In the matter of safety to men, the present boilers and machinery are more reliable than formerly, but the steam pipes and attachments, where the greatest danger lies, are really less reliable on account of high steam pressure, added to which the machinery space is so much cut up that the men cannot be kept under control of one officer. As stated before, the stresses on the main engines are less, but they come with more suddenness; and the boiler can be made just as strong, in proportion to the higher pressure, as it could thirty years ago for twenty pounds. Pipes, however, though made with a higher factor of safety, are subjected to greater variation of temperature, are held more rigidly in the many bulkheads through which they pass, and require greater care in the joints. We seldom hear of the shell of a marine boiler giving way. This accident occurred on board the *Chenango* during the war. She had box boilers and, through insufficient bracing or some carelessness in setting the braces up, the whole top blew off and struck the berth deck, a wooden deck, without lifting it out of the ship.

Longitudinal rupture of the boiler shell is not uncommon with land boilers, which are sometimes supported in such a way that they may corrode underneath the supports. If the shell of one of the large boilers of the *Newark*, for instance, parted longitudinally in a horizontal line, the pressure to lift the protective deck would be 2,640 tons, which I am sure would wreck the ship. Such an accident, however could not occur unless the

boiler were grossly neglected and the steam pressure were run far above safety. Another class of accidents, however, resulting from blowing out of tubes, overheating of furnace crowns, &c., is likely to be just as fatal to the firemen, although of no serious danger to the ship. The explosion of steam pipes is not infrequent in the merchant service or Navy. The *Tennessee* in 1883, and the *Concord* in 1891, are recent cases which have resulted fatally.

The banding of steam pipes, the practice now followed by the Bureau, is a precaution against a similar accident.

Whatever may be the cause, weakness of parts, local imperfections, or high pressures, the danger to the men is just the same and panic is sure to follow an explosion, especially if the compartment be closed. Only the men who have been in the fire room at the time of the bursting of even a small steam pipe can realize the confusion resulting. I have seen an open fire room cleared in a few seconds, and have had to lie down close to the floor plates myself to keep from being suffocated, upon the bursting of a 1-inch drain pipe from the steam drum. I do not by any means infer that we cannot find men to do the work in spite of the element of danger, but I do wish to call attention to the necessity of careful training in the work required of them, and under conditions similar to those of actual service. One thing to be said in favor of the present arrangement of water-tight bulkheads is that the effects of disaster are localized. While an explosion may be more fatal to the men inside of the closed compartments, it is not likely to spread to other compartments, and thus the machinery is not totally disabled. In the case of the *Chenango* above referred to, all the men in the engine and fire rooms were killed. On the *Concord*, the steam from the explosion of the steam pipe was confined to the wing passage and forward boiler compartment, and two men, the only ones in the compartment, were killed. The water-tight subdivision, with all its disadvantages of communication, is, therefore, both a safeguard against torpedoes and shot, and a real protection to the engineer's force, as a whole, against their own machinery when working at high powers.

I am convinced that the lack of ready communication will be seriously felt in the next war.

PERSONNEL.

I come now to the question Personnel, which officers ought to look at without prejudice, but which, nevertheless, has made a great deal of controversy in the service. In what I have to say on the subject, I have no desire to enter this field of controversy, but will confine myself to a statement of those things which I consider necessary to make the engineer's force most efficient on board ship. In his book on "The Influence of Sea Power upon History," the President of this Institution has described fully the "weather gage" which confers the power of giving or refusing battle at will and the choice in the method of attack. This unquestionably belongs to the fastest ship; and speed, in its turn, belongs to the ship with the best machinery and the best trained fire room force. All the drilling at the guns in the world will not add to or subtract from the ship's speed by one knot, while every day's training of the firemen will tell in the increased endurance of the machinery and the steadiness with which the speed can be maintained. At present, it takes from six months to a year after a new ship goes into commission to get the best results out of her machinery; and through no fault of the firemen. It is only that they do not know when first taken on board. I speak from experience, and not from general theoretical guess work. When the *Charleston* went into commission we received a green crew, composed mainly of Pacific Coast men and old-time men-of-war men. Most of those in the fire room were well intentioned, but the work required to train them was thoroughly disheartening. We started out under full power on our first voyage from San Francisco to Honolulu, with the intention of making a record, and came down to half power within two days. In the first place, the men were not accustomed to the fires, and in the next, they were unable to get the coal down from the upper bunkers rapidly enough to supply the fires. After one year's commission, during which we ran only a moderate amount, the ship made the same run under full power when

carrying King Kalakaua's body back to Honolulu, and we broke the record for long distance steaming in our own navy or any other.

With better preliminary training and greater inducement to remain in the service, I think we should have been able to obtain men who could have added another knot to the speed. And this will be the history of every new commission of our ships until some system of thorough training for the firemen is adopted—apparent failure for six months, and then very satisfactory results.

I can speak feelingly, because I have been thoroughly disheartened many times at the sight of barbarians, without knowledge or training, coming into the engine room to take care of fine machinery that cost half a million dollars and years of labor. We get some good men, undoubtedly; otherwise, the ships could not be run at all; but they seldom stay more than one cruise. It is usually the old-time "Jacky" who ships over again. This applies to the deck force as well, and I do not think the engineers have a monopoly of worry; but I do feel certain of the great number of desertions from the fire room; also, the difficulty of obtaining men. We shipped enough firemen and coal heavers on the *Charleston* to fill her complement three times during the two years I was on board. A new lot would come on board from the receiving ship at Mare Island, clean a few double bottoms and a boiler, and desert in the first port.

The percentage of desertion from the engineer's division was far in excess of the gun divisions, chargeable, I think, to the following causes:

First. Many of them come from a class who have learned trades more or less indifferently, and who do not take kindly to the discipline of board a man-of-war.

Second. Many are recruited from a class of adventurers who like frequent change and would leave under the best of circumstances.

Third. Many, if not all of them, feel like "the under dog." This is partly the fault of the commanding officer, who naturally pays more attention to the men who come more immediately beneath his notice. A little study of the firemen's needs would cor-

rect much of the trouble with them. There is a strong feeling among the men that they have too many masters. Part of this follows from the necessity of a head and discipline on a man-of-war, but I think most of it could be allayed by entrusting more of the management of the men below to the Chief Engineer. It should be just as much a regulation to give the men liberty, orders, and various duties which arise, through the Chief Engineer, as it is to forward all official papers through the Captain. Of course, I recognize, as I stated above, the necessity of military organization, which may be harassing to the firemen, but I am sure the harassment would be less in the hands of one man than in the hands of half a dozen.

Fourth. Much of the work in a steel ship is carried on in darkness and dampness, and the men really cannot stand it for long periods. This kind of work has increased enormously in the new Navy, and the improvement for the men lies in one of three directions, (1) increase in number of second-class firemen and coal-heavers, (2) extra pay for work in the double bottoms, (3) detail of men from deck to assist, so that all may partake alike of the double bottoms.

Fifth. Their treatment on board ship. Of this I shall say little, as it depends much on the personal character of the officers. All the rules in the world cannot teach practical sense and good temper.

Sixth. Lack of training, so that many of them become discouraged before they have learned to do their work to the best advantage. Some ship should be detailed for the training of firemen, where they might undergo the preliminary instruction which now has to be given during the first year of commission. I see no reason why Coaster's Island could not be made a training school for all classes of men, as well as for the apprentices who learn how to reef topsails and fire smooth-bore guns. One modern ship, preferably a coast defense or battle ship, could be very usefully employed here for both firemen and deck force. The workshop on shore could be extended and more boilers added, especially to teach the younger men those innumerable details about repairs and firing which can be better taught in a large

place on shore than in a closed and dark fire room. There is not one fireman in three who can manage a fire properly, and it is almost impossible to teach men to fire on board ship, where the attention of an engineer cannot be given entirely to that. This system could be worked out into a very satisfactory one, and, no doubt, would be under the superintendence of Captain Bunce, if the Navy Department could be induced to try it. I leave all matters of elaboration to a more fitting time, as this lecture is devoted mainly to our work under present conditions.

All line officers to whom I have talked on the subject of mechanical training recognize the fact that even the gun captains should receive a large amount of it. In these days, when young men have to go through a course at the torpedo station to fit them to run a small dynamo engine developing twenty-horse power, it is perfect folly to think about getting untrained men, even though they be mechanics, to run from forty to sixty engines aggregating from 5,000 to 20,000 horse power. Better training would necessarily be followed by greater permanency and better treatment of the men, and, perhaps, a better pay table, as they would be far more valuable to the service than the typical Irishman who now wields the oil cup and shovel.

In the matter of drills and military training of firemen on board ship, I do not think they are productive of much good. They add to the labor of the men, interfere with their work, and take them out of the hands of their own officers, and in every way tend to disorganize the fire room. By this I do not wish to be understood to decry the value of military training in its power of discipline and organization; but I think that all the drills for mechanics and firemen should be in the training school before they go to a commissioned ship, or on the receiving ship. There is no doubt that cases might occur in actual warfare when the firemen could render efficient service on deck, and, for that reason alone, I would heartily second any effort to make them well acquainted with the modern guns, but it is too much to expect of mechanics that they should take an interest in such things when they have plenty of work in their own department.

The detail of firemen and coal heavers to the powder division

at general quarters will undoubtedly decrease the efficiency of the ship as a whole. It may get the powder to the guns in a shorter time but at the expense of the manœuvring qualities and the water-tight subdivision of the ship. On the *Charleston* it took the entire division of eighty-two men to close the water-tight doors, steam valves on deck, and get everything ready for action below, and yet one-third of them were required to go to the powder division at the call to general quarters. I realize the necessity of getting powder to the guns, also of the superiority of the firemen to Chinese servants and idlers, but I cannot see the philosophy of weakening a most important part of the ship for that purpose. I know what has been said from time immemorial about the ship being nothing but a gun carriage, and I recognize its truth where she cannot be used as a ram, or does not find it prudent to run away from superior force; but I also believe that it is usually an injury to the gun carriage which puts the gun out of commission, and I am sure that the management of this enormous carriage will be far more important in the next war than even the assurance of powder supply. I need hardly say that the above remarks do not spring from any fractious spirit of opposition, but from a sincere desire to see our ships as efficient in all their parts as possible, and nobody would render more cheerful obedience than I do to all such orders as may be necessary in emergency. I am sure the Engineers would rather see the whole Engineer's force detailed to the powder division, or any other division, than to see the flag dishonored.

One of the main reasons why the parts below deck have been lost sight of is that inspecting officers usually examine the battery carefully without much thought of the machinery and the water-tight subdivisions. Those parts which are seen naturally receive the most attention. Another thing, the engines are generally under half power during inspection, or there is no steam at all. I fully believe that one serious clearance for action, with steam on all boilers and everything ready for running, would clear up this subject wonderfully. Such an inspection should be made of all our new ships, and the detail of men made in accordance with that.

Although I have dwelt on the subject of the most economical speed, in a former lecture, there are many cases where the actual cost of the coal does not measure the greatest economy in the long run. On one occasion the firemen of the *Esmeralda* are said to have bolted from the fire rooms during a full-power trial. I have no doubt this might occur in any Navy where the men had not been trained at high speeds. It only serves to emphasize the necessity of running at high speeds occasionally. Where a run is short and the coal limit is not approached, it would be an excellent plan to run full speed at least once a quarter. The additional expense would probably not be greater than that for target practice.

In this connection, consider the condition a ship would be in after a fight, even if successful, and the number of mechanics required to repair her or the prize. All mechanical work, even repairs to the guns, would fall to the Engineer's division, as it should, they being the only trained mechanics on board; so that the ship with the best machinists and firemen would be the soonest ready to run or fight again. This also applies to all those derangements incident to continuous cruising on a foreign station, even in time of peace. I cannot do better than to mention the *Charleston* and her chase of the *Itata* once more.

The run to Iquique from San Francisco was made in twenty-five days, stopping at San Pedro, Acapulco, Callao and Arica. During this time, the machinery, boilers and firemen were not spared; and numerous small defects developed and were repaired. An officer fitly designated her on the voyage as a steam collier; for we did nothing but coal ship and run. The gun divisions were constantly drilled at the guns and other exercises in preparation for any emergency that might arrive. We found the *Itata* in Iquique, practically disabled, but surrendered to Admiral McCann, who wished to get her out as soon as possible. Fifty machinists, oilers and firemen were taken on board from the *Baltimore* and *San Francisco*, and all parts of the engines, boilers and auxiliaries thoroughly overhauled. Only half a dozen men could be spared from the *Charleston* for this work, which took five days. In the meantime, the whole Engineer's

force on the *Charleston* were busy cleaning boilers and engines and repairing for the return trip. Two or three broken eccentric straps of the blowers were repaired, and a number of boiler tubes plugged, while the blacksmiths were kept busy making grate bars into supports for the battle doors over the engine room hatches. The meeting with the *Esmeralda* off Acapulco had impressed us very forcibly with the fact that we could not stay below with these doors closed, and the above supports were intended to hold the doors up at an angle of 30°, with an opening of 18 inches. I have only to ask any officer the relative importance to machinists and firemen of thorough mechanical training and drills with the guns in such case as this.

No reference has been made to the different ratings and the different kinds of hand work required below. The machinists, of course, must have a good knowledge of shop work, with some experience at sea, while the oilers and first-class firemen must be handy at anything. They must be able, among their other accomplishments, to overhaul and clean all the parts of the main engines, to run a small engine, make joints, repack large and small stuffing boxes, wrap leaky joints, lay brick, cement the double bottoms, paint, do a little carpentering, stow coal, and take charge of the plant in the auxiliary boiler room, or the steam launches.

The second-class firemen and coal heavers have, generally speaking, the dirty work of the department, such as cleaning furnaces, tubes, boilers, bilges, double bottoms, and stowing and passing coal. They constitute over half the engineer force, as they have nearly all the disagreeable work of the ship to do. They are always the most troublesome to the engineers. The greatest percentage of desertions take place from their ranks, and all the details for cleaning outside of the department have to be made from their numbers. They should not be expected to work on deck stations, excepting where there is a piece of machinery to look after. There is always more or less friction when they have hatches and ventilators on the quarter deck to clean, not to speak of the weakening of the engineer's force resulting from such duties. Besides, it is out of the question for

coal heavers to keep deck stations clean at sea. Another matter which would probably improve the whole force would be to take all the men on board ship as coal heavers, excepting such as have shown themselves efficient and well behaved on previous cruises. This will give opportunity of rewards for good service and punishments for bad by disrating for incompetency, without the tedious process of court martial.

The remarks of the Engineer-in-Chief in his last report strike me as so good that I have said little about the machinists herefore, and now content myself with quoting from him.

"Very great difficulty has been experienced in securing competent men to serve as machinists in our ships. In one case, where a vessel was fitted out in one of our largest manufacturing cities, it was believed that the securing of the complement would be aided by permitting the Chief Engineer to select his men and satisfy himself as to their competence before sending them to the receiving ship for enlistment, but though the Bureau of Navigation offered every facility for such enlistment, only one or two men were secured out of a complement of eight. It is not altogether difficult to understand the reason for this; the pay of a machinist in the Navy is fairly good, and is about what all except leading men receive in the best shops on shore. It must be remembered, however, that on shore the mechanic works only a certain number of hours each day for his pay; if he works more it is overtime, and paid for at a higher rate; after his day's work is over he is his own master and can go and come as he pleases; if dissatisfied with his employment or employer he can leave at will, conditions that are, of course, utterly impossible on board ship. On shipboard, he is liable to be called at any time when off watch, day or night, for "odd jobs" or repair work, since the men on watch can not be called from their regular duties for this. It needs but a moment's thought to realize how frequent these calls are likely to be on a vessel that has, besides the main engines, scores of auxiliaries and almost innumerable mechanical appliances scattered all over the ship; and that, owing to the small number of skilled mechanics allowed, the labor each one will be obliged to perform will be arduous in the extreme.

Then, again, the mechanic on shore does most of his work in a shop, where he has light and air and room, besides all sorts of conveniences in the way of tools; on board ship he performs much of his work in a compartment where he often can not stand erect, where the air is foul and the light (always artificial) is sometimes electric, but oftener from a smoky oil lamp. On shore he sleeps in a bed and goes to it when inclined; on ship he sleeps in a hammock and gets into it when told." * * *

"Another reason for the difficulty is the very unsatisfactory conditions as to accommodations on board ship, which have hitherto prevailed. It is gratifying to note that in recent designs for large war ships the Bureau of Construction and Repair has endeavored to remedy this defect, and I believe that if it were generally known that on our new ships the higher petty officers will have excellent accommodations we would have less trouble to secure the men we need.

"In a previous report, I spoke of another possible reason for this difficulty of securing good men for these positions as arising from the entire lack of distinction in the treatment given to these men and to the firemen and coal heavers. In all modern ships a considerable amount of responsibility must fall upon the higher petty officers, and they are required to exercise authority over the enlisted men. Still, they have been almost invariably treated hitherto, as far as accommodations and privileges are concerned, the same as the latter. It is almost impossible under these circumstances for them to enforce their authority, and, after one cruise, very few of them enlist for a second. I consider that a marked distinction should be made between the higher petty officers and the men who are subject to their orders. As far as possible, I think they should be treated more as non-commissioned officers in regard to permission to go on shore and similar privileges. If they have a place to themselves in the ship and are allowed leave on shore when their services can be spared without having their names placed on the general liberty list I believe they will take pride in their positions, will recognize that they are considered important by the Department, and will, consequently, live up to their privileges. In any case I believe the

experiment worth trying. If it fails, we shall be no worse off than we are now, and, if it succeeds, we shall have settled a question which is now proving a serious embarrassment."

The above also applies to some of the ratings on deck, and much good would follow a thorough overhauling of the regulations with regard to chief petty officers.

STEAM LAUNCHES.

The steam launches of our service have been much maligned, and, while they are not of the most advanced type, they are generally as satisfactory as any steam boat would be under the treatment they receive. With the Ward boiler and the keel condenser, I see no good reason for criticism of the machinery. I believe much of our trouble heretofore has been unavoidable with the irregularity of overhauling and the hard treatment. Where a ship has two steam boats, each one should be laid off for two or three days at regular intervals to receive thorough cleaning and repairs.

THE ELIMINATION OF SULPHUR FROM IRON.

A New Process for the Purification of Iron and Steel from Sulphur.*

By E. H. Saniter, F. C. S.

In September, 1890, I commenced a series of experiments with a view of removing the sulphur from iron.

I was impelled to this research by the recognition of the fact that sulphur was the worst enemy, and the only one over which the iron and steel manufacturer had not as yet obtained complete control. With this fact in view, I made some experiments to ascertain the effect of prolonged contact of lime with sulphury iron at a high temperature. The results obtained were of an irregular and imperfect description. Reasoning from these results, I came to the conclusion that a more readily decomposable body than lime was necessary for the rapid removal of sulphur from iron. Recognizing the fact that chloride of aluminium and other chlorides are readily reducible to the metallic state, I determined to try if calcium chloride, acting in this direction, might not act upon sulphide of iron more readily. Whether calcium is produced or not, I soon discovered that calcium chloride and lime, *i. e.*, the oxychloride, is a very powerful desulphurizing reagent.

Number of experiment.	Time during which the molten iron was in contact with the mixture.	Sulphur in Iron.		Mixture Used.
		Before Treatment.	After Treatment.	
	hours.	*per cent.*	*per cent.*	
4	2¾	.38	.03	Lime.
6	1¾	.30	.12	Lime.
15	½	.42	trace	Lime, 90 per cent.; calcium chloride, 10 per cent.
16	½	.42	"	Lime, 90 per cent.; calcium chloride, 10 per cent.

* Paper read before the Iron and Steel Institute of Great Britain.

The foregoing table indicates clearly the comparative results obtained, the sulphury iron being kept molten in plumbago crucibles in contact with the substances named.

These results show (1) that lime alone removes a considerable quantity of sulphur from iron if the contact is sufficiently prolonged. (2) That a mixture of calcium chloride and lime in the short space of half an hour completely eliminated the sulphur, Further, the lime and chloride of calcium mixture only softened but did not fuse. It is upon these last experiments that my process has been elaborated.

It consists essentially in bringing chloride of calcium and lime into contact with molten iron or steel under certain well-defined conditions. Before proceeding to describe these conditions, it will be desirable to give some information on the important subject of "chloride of calcium." The salt is produced as a by-product, to the extent of many thousands of tons, in the manufacture of ammonia, soda (by the ammonia process), and Weldon's bleaching powder process. I am informed on the very best authority that at the present time not more than 10 per cent. of this large quantity is utilized, the remainder running to waste for want of a market.

The driest calcium chloride obtainable at present contains 70 per cent. calcium chloride and 30 per cent. of water, and costs about £3 a ton. Before using, it is generally desirable to dry it and this may easily and cheaply be accomplished in a reverberatory furnace at a low heat, the whole of the water of calcium chloride being driven off at a temperature of 220 degrees Fahr. Calcium chloride may also be cheaply produced by mixing "waste pickle" or similar liquors with lime, and utilizing the mixture of calcium chloride, oxide of iron and lime in the basic open-hearth furnace. Fluor spar in conjunction with lime has considerable desulphurizing properties, but it has also some considerable disadvantages, viz., its comparative infusibility, and the severe cutting action which it has on the lining when used in furnaces lined with "basic material."

I now come to the description of the process as applied to the removal of sulphur from raw or cast iron. A mixture of cal-

cium chloride and lime is prepared, which will fuse readily at the temperature of the iron to be operated upon. The desired combination is made by grinding calcium chloride and lime together in a mill to thoroughly mix them, and also to bring them to a moderately fine powder. About equal parts of each are required to give the desired fusibility. This mixture is then placed on the bottom of a ladle or receiver, and consolidated by heat or kept in position by other suitable means. The heat may be applied in the first instance by means of a blowpipe arrangement using blast furnace gas, but when in continuous use the heat of the ladle itself is quite sufficient. The receiver is then filled with iron, which may be drawn direct from the blast furnace, the heat of which melts the mixture, which, rising up through the metal, removes the sulphur very completely.

I do not find it necessary to have a reducing atmosphere, and, indeed, oxidation may be going on concurrently with the removal of the sulphur, as will be seen later on. Notwithstanding this, however, the sulphur is removed as sulphide. Should it be desirable to remove silicon, as well as sulphur, the lime of the mixture is replaced by hydrate or carbonate of lime, or even oxide of iron in addition, should these first be insufficient. About 25 pounds of chloride of calcium and an equal weight of lime per ton of iron have been found sufficient to effect purification. In the many trials made, about three tons of iron direct from a blast furnace were treated in a ladle at each operation.

Although the first or coldest iron from the furnace and hydrate of lime in the mixture were used, no "skull" was made, which is a fact of the very greatest importance, and there can be no doubt that if such a small quantity of metal is not materially chilled we may safely assume that larger masses would never "skull." The uniformity of the results obtained was very marked, as will be seen by the accompanying analyses.

Nos. 7 to 13 are consecutive charges, and show the regularity of the results:

Number.	Class of Iron.	Sulphur.		Silicon.		Mixture Used.
		Before.	After.	Before.	After.	
		per ct.	*per ct.*	*per ct.*	*per ct.*	
1	No. 5 hematite.....	.220	.060	1.6	1.2	Chloride of calcium and hydrate of lime.
2	Hard forge, 1.5 per cent. phosphorus.	.300	.060	1.7	1.6	"
3	Grey forge	.070	.008	2.2	1.9	"
4	Basic iron	.197	.072	Not estimated.		"
5	"	.191	.062	"		"
6	"	.109	.031	"		"
7	"	.102	.032	.56	.32	"
8	"	.065	.026	.84	.46	"
9	"	.093	.016	.32	.09	"
10	"	.089	.024	.42	.18	"
11	"	.083	.020	.37	.09	"
12	"	.133	.030	.70	.32	"
13	"	.091	.026	Not estimated.		Chloride of calcium and limestone.
14	"	.060	.008	"		"

It should be stated that the results obtained above were from the application of the process when it was well known from the appearance of the cinder that the resulting pig would be abnormally high in sulphur. The above table shows an average elimination of:

	Per cent.
1. Sulphur, ...	73.6
2. Silicon, ...	35.77

The removal of silicon being due to hydrate of lime only. The following is an average slag produced by the above treatment:

	Per cent.
Calcium chloride, ...	39.1
" sulphide, ...	5.8
Lime, ...	38.6
Silica, ...	12.9

A considerable part of the chloride of calcium in this slag may be dissolved out with water, and recovered for future use.

Only a limited quantity of iron (about 50 tons) has been treated in this way owing to proper plant not being as yet available, the small 4-ton ladle being only a make-shift. The ladle was lined with ordinary fire bricks, which were practically unattacked by the slag at the comparatively low temperature at which the process was conducted.

Appliances are in course of construction which will deal with the whole output of the furnace as the metal is run. The plant required is of a simple and inexpensive character, consisting of ladles or receivers on wheels. The cost of materials at present prices is about 6d. per ton of iron treated, and this will be less when a more efficient receiver is used. It is also very probable that should a demand arise for chloride of calcium the price would go down. Against this extra cost may be set the cheaper production and enhanced price of the pig iron produced. This process can be adapted to a considerable number of uses, such as: (1) The purification of hematite, basic and common (1.5 per cent. P.) irons as they run from the blast furnance or cupola, thus producing these qualities of iron low in sulphur and silicon, after which they might be used for direct steel making or cast into pigs. (2) The purification of steel in the ladle, after it leaves the furnace or converter.

It is a fact pretty well known and established that no sulphur is eliminated in the "basic open-hearth" process as ordinarily worked, and that not only is this the case, but when ore containing much sulphur is used for feeding, the bath of steel takes up sulphur, so that, under these circumstances, it may contain twice as much sulphur as that in the pig and scrap originally used. This is shown by Wedding ("Journal of the Iron and Steel Institute," page 547, Part II, '90), and is confirmed by my own experience. In my process, however, as applied to the basic open hearth, sulphury iron and mineral may be used, and not only is the sulphur not increased in the steel, but a very considerable elimination takes place.

In order to attain this result, it is necessary, at an early period after the charge is melted, to obtain an exceedingly basic slag, and to add a suitable quantity of calcium chloride to it. By a

very basic slag I do not mean what has hitherto been considered as such, but a step in advance of that with about 50 to 60 per cent. of lime. If these conditions be obtained and maintained, it will be found that sulphur is eliminated along with the carbon and phosphorus, and in as satisfactory a manner.

The best method of obtaining this condition of slag is to charge along with the metal and scrap a much larger proportion of limestone than usual, about two cwt. to the ton. When the charge is melted the slag will be of the required composition, and the chloride of calcium may then be added in several lots at intervals. The quantity of 70 per cent. chloride used is slightly under ½ cwt. on the ton of ingots made. The following table shows the quality of iron which has been used and the steel made from it:

	Pig iron used.				Average per cent. of sulphur in metals charged.	Steel made.				
No.	Si.	S.	P.	Mn.		C.	Si.	S.	P.	Mn.
	per ct.	*per ct.*	*per ct.*	*per ct.*		*per ct.*		*per ct.*	*per ct.*	*pr.ct.*
1	.04	.76	1.3	.18	.58	.215	trace	.081	.027	.68
2	.10	.45	2.1	.50	.35	.20	"	.072	.052	.75
3	.10	.45	2.1	.50	.35	.19	"	.048	.054	.59
4	.04	.25	2.6	1.00	.20	.08	"	.048	.025	.43
5	.20	.23	2.6	1.00	.19	.17	"	.048	.045	.57
6	.20	.22	2.6	1.00	.18	.145	"	.063	.042	.73
7	.20	.22	2.6	1.00	.18	.39	"	.053	.045	.61
8	.40	.22	2.5	1.30	.18	.18	"	.018	.034	.40
9	.20	.17	2.6	1.20	.14	.15	"	.038	.040	.58
10	.18	.16	3.1	1.50	.13	.13	"	.032	.040	.20
11	.44	.15	3.5	1.50	.13	.75	"	.042	.040	.60
12	.20	.15	2.6	1.00	.13	.15	"	.038	.040	.58
13	.65	.13	3.3	1.50	.11	.155	"	.025	.035	.68
14	.56	.05	.05	3.00	.05	.115	"	.016	.010	.12

The pig used in the above charges was 75.0 per cent. These analyses have been selected to show the varying percentages of sulphur in the cast iron used, and the different grades of steel made. No. 10 is conductivity steel. No. 14 made from white hematite is Swedish bar quality.

The Wigan Coal and Iron Company have now manufactured over 2,000 tons of steel, of which the above are examples, from sulphury iron, the process being continually and successfully

worked by them. The steel has been sold for all purposes for which basic open-hearth steel is used, namely, wire, hoops, rivet steel, tin bars, &c., and has been found to be fully equal in quality to that made from pure cast irons.

It is evident from what has been said that no great care is necessary in the selection of material, the only objectionable elements being silica and silicon. The commonest descriptions of iron scrap and ore may be used subject to the above reservation. The yield of ingots obtained is as good as that in the use of low sulphur iron. The use of common iron high in sulphur and low in silicon and carbon has the advantage that a less quantity of steel scrap is required, and that the consumption of ore for feeding is reduced very considerably.

The cost of the chloride of calcium is about 1s. per ton on the weight of ingots, but owing to the saving effected in the cost of material and in the quantities of scrap and ore used, there is in reality a saving of about 4s. per ton of ingots. Neither the hearth nor brick work of the furnaces, nor the regenerators are in any way affected by the use of calcium chloride, as has been proved by practical experience over a period of six months.

As it is desirable in case of anything new to have corroborative evidence, Mr. Stead on two separate occasions visited the works of the Wigan Coal and Iron Company and investigated this process. The results of his investigation confirm what I have stated. I understand Mr. Stead's investigations will be embodied in the paper which he is reading on "Sulphur and Iron." The process has been patented in this and other countries.

I cannot conclude this paper without recording my sincerest thanks to Mr. W. H. Hewlett and Mr. T. M. Percy, of the Wigan Coal and Iron Company, who by their enterprise and by daily advice and encouragement have contributed very largely to the attainment of the results laid before you to-day. I have also to thank Mr. Stead (my teacher in chemistry) for much friendly and valuable assistance.

BENDING TESTS AND CONTRACTION OF AREA AS PROOFS OF QUALITY.

By P. Kreuzpointner.*

(Reprinted from "Engineering News," May 12, 1892.)

It is a peculiarity of soft steel having under 70,000 pounds per square inch tensile strength, to show, in the tensile test, little or none of the effect of having been burned. The writer has burned soft steel to scintillation, so that it was rough and "lumpy" on the surface, and yet the metal would have filled ordinary specifications. This is still more apt to be the case if the metal is burned while being rolled and receives more or less work afterwards. The effects of burning are, of course, modified or aggravated by the chemical composition of the steel.

While overheated steel which stands the tensile test may not be dangerously defective in certain positions, yet there is a risk attached to its use; and if that risk can be avoided it is better for the engineer and the steelmaker. Burned steel has its elasticity and density impaired. This is more or less restored by the slow stretching of the metal under tensile test, which is like so much additional work put on the metal; hence the apparently satisfactory results of such burned metal in tensile test.

The engineer finds a means to discover burned metal in the bending and nicking test, especially the latter. Prof. Krohn is, therefore, perfectly right when he says:

"Bending tests give us the surest method to discover burned metal. If the test piece is previously nicked it will show its brittleness by breaking readily after bending it to a comparatively small angle, and the fracture will reveal the characteristic coarse-grained burned structure."

The nicked bending test is also very useful in determining

* A paper suggested by reading Prof. Krohn's investigation of soft steel ("Eng. News," Jan. 9–16, 1892.) and some of the comments thereon.

the variations in quality of good structural metal. Aside from the question of injury by burning, there are degrees of excellence in metal of nominally the same grade. In a nicked bending test the best metal will bend to the greatest angle before breaking.

The wise engineer will sometimes have occasion to take advantage of this simple means of discrimination and will use the best parts of a shipment for the more important portions of his structure. Such nicked bending tests may be conveniently made in the vise, or by holding the piece tight between the anvil and the piston block of the steam hammer and striking it with a hammer or light sledge. After the first or second blow the metal will perceptibly stiffen as the next few blows are delivered. Afterward, the resistance remains constant until the piece breaks. The metal which is not quite so good will break with a less angle of bend at the beginning of this stiffening period.

An expert can thus, in an emergency, determine the quality of a metal pretty closely. Even the elongation can be determined approximately by a bending test, without nicking the test piece. If the datum points or punch marks are laid out on the straight piece and it is then doubled, without hammering it flat, so as not to disturb the marks, the distance measured around the convex surface of the bent piece between the original marks will approximate, and sometimes equal, the amount the piece would have elongated in the tension machine.

It is not always necessary to nick a test piece before bending, in order to discover its burned condition. On the convex surface of a bent piece of "soft" burned steel, there are visible numerous little cracks, very small and but skin deep, but sufficient to give the surface a rough appearance, which is not observed on a sound piece when bent double. But one unfamiliar with testing will find the nicked bending test a more satisfactory method of discovering burned metal. In using the latter, however, it must be remembered that soft steel will often show a granular fracture, especially if thicker than $\frac{3}{8}$ inch. The bending fracture of burned steel has a whitish hue; the crystals shine bright and fiery and show distinct, well-defined faces,

whether large or small, and the granular or crystalline appearance of the fracture is very marked and coarse.

The nicked bending fracture of soft steel not burned will have a bluish gray hue, with the structure not sharply defined or even "mushy" in appearance, though otherwise "clean" and free from defects.

The fracture of a nicked bending test is really a safer index of the structure of a metal than the fractured surface of a tensile test piece. In the latter case, all evidence of the original condition—that is, of the condition of the metal before it was subjected to stress—is changed or entirely obliterated, through the strain exerted on the metal at the point of fracture, which is like so much additional work put on the metal by roll or hammer. The force necessary to pull the piece apart is generally sufficient to change the structure at the point of fracture from coarse to fine, or from fine to amorphous or silky.

A few words now on contraction of area at the point of fracture, as an index of the quality of the metal. In order not to be misunderstood, the writer would emphasize the statement that to the expert in the physical laboratory contraction is a valuable indicator of the qualities of a metal, and in special cases, when very soft steel or iron is under consideration, it is even a practical guide to determine the suitableness of the material for a given purpose.

But with these qualifications ends the everyday practical usefulness of contraction. The fact is that contraction retains a certain amount of importance by virtue of other measures which accompany it. If left to stand on its own merits, it is a failure; and it proved a failure in the old German Government specification, where strength plus contraction was the prescribed standard measure of quality. Even Woehler, who was the father of the contraction theory, could not secure another official recognition of his pet child.

The crucial way to ascertain the value of a test for iron and steel is the ability to determine by it, and with it alone and without the additional help of other tests, the main characteristics of a given metal for a given use with fair accuracy.

In the elastic limit alone we have a pretty safe indication of what the metal will endure in service. The ultimate strength, taken by itself, is certainly useful to the designer. The elongation, being proportioned to the resilience, gives us an indication of that characteristic of a metal, but at the same time is a reliable measure of ductility in all parts of the metal under stress, and as such is a trusty exponent of excessive softness or brittleness. The nicked bending test not only reveals defects in structure or homogeneity, but in case of emergency can be made to do duty for the more elaborate tensile test.

But what value has contraction of area by itself if we should happen to know nothing else of the metal? Whoever was able to establish the proportionality of contraction to strength or durability? No use can be made of it as an emergency test, because before contraction can be measured the most important qualities of elastic and ultimate strength and ductility have already been determined. As an additional measure for ductility it is of no use to those who recognize elongation as a ductility measure. If elongation is a standard measure for ductility, what do we need another less reliable standard for? To check off a more reliable standard?

Again, is it rational, scientific, or in the interests of safety and economy, to adopt a test as a standard which cannot be accurately made? Even ardent admirers of contraction admit that no two men can measure contraction alike, nor can the same man measure contraction alike twice in succession. Elongation presents no such difficulties, nor does it indicate local conditions only in the metal, which is a fault of contraction. If the metal is burned, elongation and other tests will show it before contraction. If the metal is highly heated only, contraction will probably be excessive, and thus will show an exaggerated value of the metal which it does not possess. Hence, a double standard for measuring ducility is an absurdity, which does mighty little good in specifications, but is liable to do a great deal of harm in more than one way.

The writer knows of cases where really first-class steel was rejected because of little contraction, and second-class metal was

accepted. Thus, the engineer cheated himself and forced the manufacturer to furnish a poorer grade of material against his will. Excessive contraction indicates excessive softness, and metal excessively soft is wanting in the elasticity and resilience which enables a metal to withstand shock and vibration.

The relation between the percentage of contraction and the limit of elasticity is something which no engineer ever attempted to express mathematically. Hence when a minimum percentage of contraction is specified there is no assurance that the metal, while very ductile, may not be objectionably low in elastic strength.

The deceptive nature of contraction as a test of the quality of metal is well shown by the two test pieces illustrated herewith in Fig. 1 and Fig. 2. Lines and circles were drawn on the flat surface of each test piece before and after test. The ellipses in the figure are the circles drawn before test, which were elongated to their present form by the stretching of the metal. The distance between the circles and the ellipses, therefore, shows the elongation and flow of the metal at different points of the test piece. The original width of each test piece is shown at the extreme left, where the grip marks are.

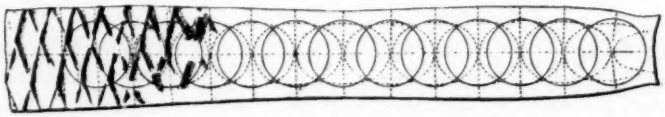


Fig. 1.

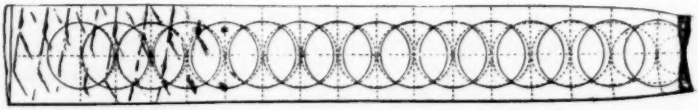


Fig. 2.

TENSILE TESTS OF STEEL.

Comparing the results shown by these pieces we find that the metal in Fig. 2 had 5,000 pounds higher elastic limit and a proportionately greater ultimate strength. The elongation in Fig. 1 is greater by 6 per cent., and contraction is greater by 12 per

cent., than in Fig. 2. Measured by Prof. Krohn's arbitrary standard of value, the metal in Fig. 2 would be rejected, having but 38 per cent. contraction instead of the prescribed 40 per cent.

This shows at once the absurdity of setting up contraction as a test for everyday use. The metal which possesses the desirable quality of high elasticity is to be rejected because it has 2 per cent. less contraction than is specified, a test whose results are so difficult of measurement that any one may miss it by 2 per cent. On the other hand, the metal of 5,000 pounds lower elastic limit is to be accepted because of a greater percentage of contraction at the point of fracture, which may be due to a soft spot in the metal indicating want of uniformity.

But referring again to Figs. 1 and 2, which are accurately reproduced from the original test pieces, it will be seen by the circles and ellipses that the metal in Fig. 2 stretched very regularly, and tapers down gradually to the point of fracture, where it contracts sufficiently to show a well-worked, sound metal. The metal in Fig. 1, however, stretches more at one place than another, and indicates, to the expert, rapid flow, lack of "body," and liability to easy "fatigue," which means fracture in detail and short life.

At the conference held at Dresden, Munich and Berlin to establish a standard code for testing, and with such eminent authorities as Bauchinger, Martens, Tetmajer, Goedicke, Hartig, Belelubsky, Brauns, Bach, Kerpely and Kik, it was decided that: "Only the work done by the test piece before the beginning of contraction is of any practical value." Kirsch, at the Royal Test Department in Berlin, came to the conclusion, during his researches on the flow of metals, that: "When a test piece breaks without contraction, whether with or without the forming of a 'cup,' and provided there are no such defects in the material as would naturally prevent or augment contraction, then the material possesses the highest degree of uniformity in regard to flow." This is another confirmation of the statement that the metal in Fig. 2 is superior to that in Fig. 1, though the stickler for contraction would reject it. Tetmajer, while agreeing with Kirkaldy and Woehler, that contraction measures the

ductility of the metal at the point of fracture, asks what practical value such a measure has to the engineer who needs to know only what the metal can stand within the maximum strength. He says: "Contraction and elongation at the point of rupture are of no practical value to the engineer." Hartig and Jenny agree with him. Many other authoritative statements as to the deceptive nature and lack of practical value of contraction could be cited to convince the progressive engineer that he will run no risk at all if he leaves contraction out of his specifications, and reserves this part of inquiry for special occasions and the physical laboratory.

THE ALLEN DENSE-AIR ICE MACHINE.

The Allen dense-air ice machine is the invention of Mr. Leicester Allen, who, seeing the weak points of the previously built air ice machines which took air from the atmosphere, compressed it once and expanded it back to atmospheric pressure, had an elaborate apparatus built in which consecutive compounded compressions and expansions could be obtained. From the results of experiments with this apparatus, he determined practically the most suitable stages and points of compression and expansion for a refrigerating machine, using air as a medium.

The final result was the present style of Allen dense-air ice machine, which uses the same body of air over and over again for the refrigerating medium, compressing it, cooling it by water by surface contact while compressed, and expanding it in an expansion engine down to the starting point.

Theoretically, the same total quantity of heat, as imparted by the compression, will disappear during the expansion, and as the cooling water has absorbed the heat of compression and reduced the temperature of the air down to the water temperature, the expansion, starting from that point of temperature, brings the resulting temperature of the air down to a very low point. This air, of low temperature and low pressure, is pushed by the expander piston through the conveying pipes to an ice-making chest, then to the meat closet and then to the scuttlebutt, cooling each in turn to its required temperature, it being continually contained in the pipes and doing all its refrigerating by surface contact. It then passes on to the inlet of the air compressor for another turn of compression, cooling, expansion, refrigeration and utilization.

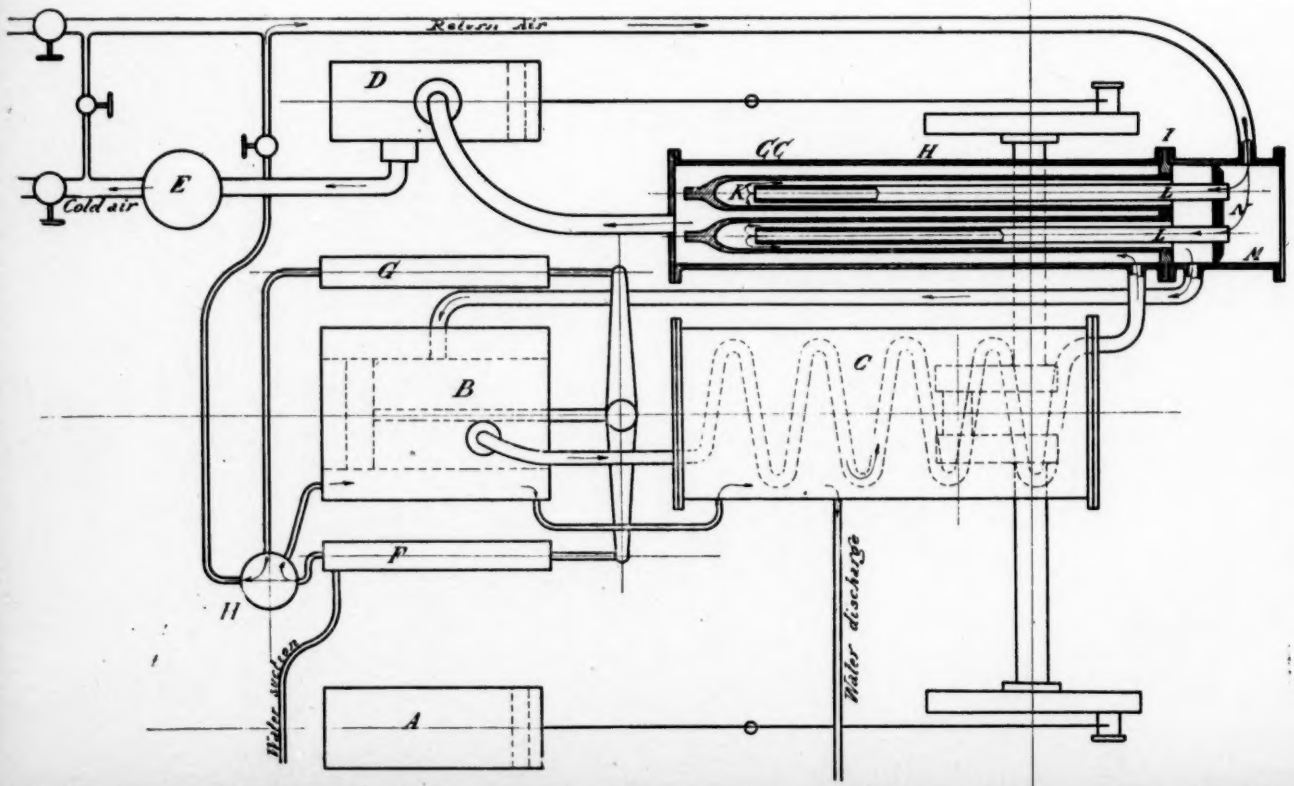
By the experimental apparatus it was demonstrated that the higher levels of comparison and expansion possessed practical advantages considerably greater than the proportion due to the

increased quantities of air handled on account of increased initial pressure. About 60 pounds (per gauge) initial pressure of air, which is then compressed to 210 pounds (per gauge), and expanded back to 60 pounds was fixed upon as a practically suitable working pressure. In order to obtain and to maintain, with a certain amount of leakage, this initial pressure, a small compressing air pump is embodied in the machine, which takes air from the atmosphere and discharges it into the initial (60 pounds) pressure, having on the way passed it through a trap, which, by cooling it, condenses and separates the atmospheric vapor contained in the air. Another trap separates the lubricating oil, which is entrained by the air from the compressor and expander engines. The steam cylinder and a circulating water pump complete the whole apparatus when used for cooling drinking water.

When the utilization of the cold air stops with the refrigeration of the meat closet, and the air, therefore, returns from its work at about 28 degrees, more or less, this cold is utilized in a special cooling apparatus to further cool the compressed air (210 pounds pressure) after it has been cooled by water. It thus goes into the expanding engine at a much lower temperature and emerges therefrom at a correspondingly much lower temperature.

The machines have thus far been made in sizes varying from 5 tons refrigerating power (equal to the refrigerating power of 5 tons of ice per day) to one-ton machines. Five-ton machines are on the United States and Brazil Mail Steamers *Segurança* and *Vigilancia*, and on the Ward Line steamer *Yucatan*. Three-ton machines are on the Pacific Mail Steamship Company's steamers *Colombia*, *Peru*, *Nicaragua*, *Costa Rica*.

The one-ton size is the only one which has as yet been used on such U. S. naval steamers as have ice machines. It is also the size which is usually employed on steam yachts, although two-ton machines have been placed on large yachts. The machine especially commends itself to the yacht and merchant services, because, while the ice machine proper is placed in the engine room under the eye of the regular Engineer force,


Return Air
D
E
Cold air
GG
H
I
K
L
N
M
G
B
C
F
H
Water supply
Water discharge
A

the cold air is conveyed in rather small pipes to different places where the ice is made and the meat is kept, sometimes over a hundred feet distant.

The annexed sectional view shows the construction of the apparatus, of which a detailed description follows:

The Allen dense-air ice machine consists of the following eight parts:

A, the steam cylinder, which furnishes the power required by the whole apparatus.

B, the air-compressor cylinder, which compresses the air to about three times the entering pressure, which compression causes the air to heat considerably. This cylinder is, therefore, surrounded by a water jacket, in order to keep the piston packings from withering.

C, a copper coil in a bath of water; the compressed hot air passing through the coil cools to the temperature of the cooling water.

CC, the return-air cooler, further cools this compressed air by means of the cold air returning from the meat house.

D, the expander cylinder, to which the cooled compressed air is admitted till it fills one-third of the volume of the cylinder. It is then shut off, and the piston continuing its passage to the end of the cylinder, the air is expanded to one-third the compressed tension (viz: to the same tension which it has when it enters the compressor). This expansion cools the air about as much as the compression heated it; therefore, the air leaves this cylinder at a very low temperature. This air is then discharged into a well insulated pipe, which conveys it to the point where the cold is to be utilized. There the pipe surface is exposed and the temperature of the surrounding water or air reduced by conduction. After this utilization the air is returned to the machine, continually contained in pipes, and it enters the compressor for a new turn of compression, cooling and expansion. Before the expanded refrigerated air goes into the conveying pipe, after leaving the expander, it passes through a trap,

E, which gathers the lubricating oil used in the cylinder, and also a slight amount of snow, leaving the air very pure and clean

when it runs through the pipes. This trap is provided with a heating pipe, and once in twelve or twenty-four hours this trap should be warmed up and the deposits be drawn off by the bottom cock. The machine is arranged, at the same time, to thaw out and blow into the trap any possible frozen deposits from the expander cylinder.

F is a common plunger water pump, which supplies the cooling water for trap *H*, for the bath around the copper coil *C*, and for the water jacket of the compressing cylinder *B*, and the machine must be stopped if the supply should fail.

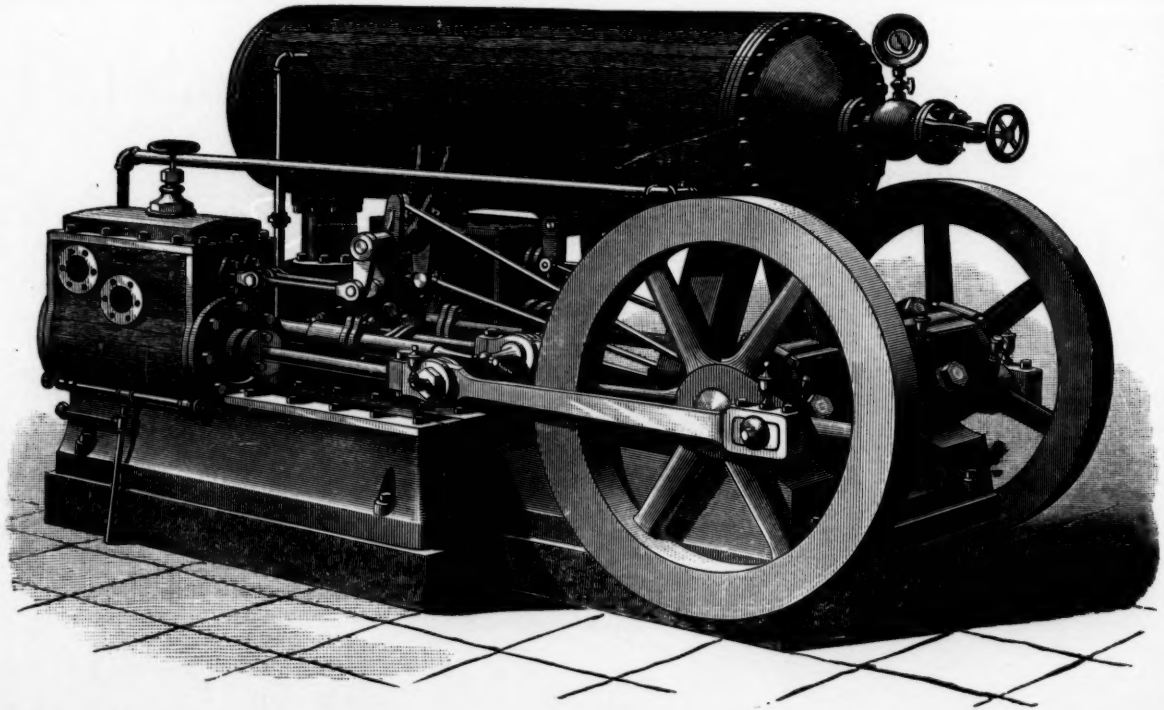
G is a small, supplementary air pump, which at starting primes the machine and the run of pipes with air to the requisite pressure, and which replaces losses of air from leaks through stuffing boxes and joints.

H is a small trap, which extracts moisture from this newly supplied air, so as to have it enter the machine as dry as possible. The gathered water must be drawn off occasionally from the bottom petcocks.

The machine is constructed to use an air pressure of 60 pounds (75 pounds total) in the conveying and refrigerating pipes, and the air compressor compresses this to 210 pounds (225 pounds total). If these pressures cannot be maintained, it is an indication that leaks have occurred, and the piston packings, and possibly the run of pipes, have to be examined. If the trap or any portion of the run of pipes is allowed to be choked by frozen oil or snow, or by closing of valves, the relation of the pressures will be disturbed. The cylinders have to be lubricated by the best quality of mineral lubricating oil, from which the paraffine has been removed by freezing. We recommend Leonard & Ellis's Extra Machine Oil. The air-compressor valves are regular engine slide valves, moved by eccentrics. The expander is also similar to a regular steam engine.

The piston packings in both of these are especially prepared leather, of which duplicates are furnished with the machine.

Particular attention is to be given to the necessity of a constant water supply, as the very life, as well as the efficiency of the machine, depends upon it.



TRIAL OF ALLEN DENSE-AIR ICE MACHINE ON U. S. S. *ATLANTA*, OCTOBER 26 AND 27, 1892.

Hour.	Steam pressure at engine of ice machine (per gauge).	Revolutions of ice machine.	Average pressure at end of compression (absolute).	Average pressure at beginning of compression (absolute).	Temperature of air discharge (Fahrenheit).	Temperature of circulating water in condenser of ice machine.	Temperature of water pumped into scuttle-butt.	Temperature of water drawn from scuttle-butt.	Pounds of water drawn from scuttle-butt.	Pounds of ice made and removed.	Thermal units removed.	Remarks.
Oct. 26.												
a. m.												Trial commenced, and ice-making cans filled, at 4·10 A. M.
4·10	55	75	230	74	45	53	72	37	0	0	0	
5·00	55	70	230	75	45	53	73	38	416.9	0	14592	
6·00	53	70	230	75	41	53	73	39	225.1	0	7653	
7·00	58	70	230	75	40	53	73	39	250.1	0	8503	
8·00	60	80	235	76	42	53	73	39	375.2	0	12755	
9·00	63	85	230	76	43	53	73	39	483.6	96	34087	At 8·30 ice made and taken out.
10·00	61	85	235	76	45	53	73	40	166.8	0	5504	
11·00	65	85	240	77	45	53	73	40	166.8	0	5504	
Noon.	55	70	220	74	44	53	73	40	200.1	0	6603	
p. m.												
1·00	60	80	238	80	41	53	73	40	400.2	88	29381	At 12·30 ice made and taken out.
2·00	62	85	240	80	41	53	73	41	333.5	0	10672	
3·00	52	80	240	80	41	53	73	41	166.8	0	5337	
4·00	60	80	220	74	41	53	73	41	416.9	0	13340	
5·00	53	65	200	65	41	53	73	43	125.1	89	20361	At 4·15 ice made and taken out.
* 6·00	...	..	...	...	...	...	...	...	...	...	...	
* 7·00	...	...	...	...	...	...	...	...	...	...	...	
* 8·00	...	...	...	...	...	...	...	...	...	...	...	
* 9·00	...	...	...	...	...	...	...	...	25.0	...	...	
10·00	58	75	218	78	30	53	73	46	83.4	0	2252	Ice-making cans filled and started engine at 9 P. M.
11·00	60	80	235	87	30	53	73	43	166.8	0	5004	
12·00	65	83	230	80	37	53	73	40	125.1	0	4128	
Oct. 27.												
a. m.												
1·00	63	86	225	76	37	53	73	40	250.1	85	22226	At 12·10 ice made and taken out.
2·00	63	85	228	78	38	53	73	40	200.1	0	6603	
3·00	62	85	230	80	38	53	73	41	325.2	0	10406	
4·00	62	85	230	80	40	53	73	40	125.1	0	4128	
5·00	62	85	220	74	38	53	73	41	233.5	84	22909	At 4·30 ice made and taken out.
6·00	60	85	220	74	39	53	73	41	166.8	0	5337	
7·00	65	85	220	74	39	53	73	41	250.1	0	8002	
8·00	63	85	220	74	39	53	73	40	250.1	0	8252	
9·00	63	85	213	70	39	53	73	42	275.2	88	24705	At 8·05 ice made and taken out.
10·00	63	85	220	74	39	53	73	40	41.7	0	1376	
11·00	63	85	215	72	39	53	73	41	166.8	0	5337	
Noon.	63	85	210	70	39	53	73	42	58.4	88	17985	At 11·30 ice made and taken out.

* Machine not working because of low pressure of steam due to demand for other auxiliaries.

As showing what one of these machines will do in every-day work, the preceding account of a test of the one on the U. S. S. *Atlanta* is interesting.

The perspective view gives a good idea of the external appearance of the machine.

The one-ton machine, as supplied to vessels of the U. S. Navy, occupies a floor space of 6 feet 4 inches, by 3 feet 0 inches, and is 3 feet 9 inches high.

The Allen machine is supplied by Mr. H. B. Roelker, 41 Maiden Lane, New York City, who has kindly supplied the foregoing cuts and description.

NOTES.

DESCRIPTION OF THE LIFTING AND HAULING APPLIANCES IN PORTSMOUTH* DOCKYARD.

By Fleet Engineer John T. Corner, R. N., Chief Engineer of the Dockyard.

[Extract from a paper read before the Institution of Mechanical Engineers, at Portsmouth, at the summer meeting, 1892.]

Compressed Air.—In the most modern part of the dockyard the lifting and hauling appliances are worked chiefly by compressed air; the only exceptions are the heavy cranes and sheers, which are worked by steam power direct. The air is compressed to 60 pounds pressure per square inch into eight wrought-iron receivers, having a total capacity of 18,000 cubic feet. The compressing is done by one or other of two separate sets of pumps. One set consists of two pairs of compressing pumps, worked through gearing, either separately or together, by a pair of simple engines of 90 I.H.P. The other set of pumps is worked by a pair of compound beam engines of 200 I.H.P. This machinery is situated at the main pumping station, about the center of the yard; and, besides the air-compressing machinery, the same building contains the main dry dock pumping machinery of 1,000 I.H.P., and two pairs of 120 H.P. engines for general fire and dock drainage purposes. The larger set of air-compressing pumps will fill the eight receivers to 60 pounds pressure in one hour. No case of a receiver bursting has occurred here; and there is no record of a pipe having been replaced during the last two years. There is also less trouble with air joints than with steam and hydraulic joints.

The air pipes, which have a total length of 14,000 feet, or about $2\frac{3}{8}$ miles, and vary from 3 to 12 inches diameter, extend round the large basins, and are connected to forty 7-ton capstans, to five 20-ton cranes, and to the machinery for working seven caissons and numerous penstocks, besides driving a small workshop engine. The air pressure is also used occasion-

* England.

ally for driving small engines for carrying out machine work on board ships building; and it has also been connected with the auxiliary steam pipes of some of the larger battle ships, so that air pressure could be used instead of steam for driving the hydraulic pumping engines on board the ships for working the gun gear for drill purposes, and also for driving the electric light engines and other auxiliary machinery on board. This obviates the necessity of getting up steam in the ships' boilers, thus admitting of their being kept systematically in a certain condition, either closed and dry, or quite full of water, which would be impractical if they were being used at irregular intervals and at short notice.

Steam Power.—The direct use of steam for the lifting appliances is almost entirely confined to the heavy cranes and sheers, to the portable machines, and to a few cases where there is steam power conveniently near. The large tripod sheers at the east end of the repairing basin, by Messrs. James Taylor & Co., of Birkenhead, have a working capacity of 80 tons, and have been tested to 120 tons; their height is 140 feet, and they can overhang the basin wall 40 feet, the back leg being worked in and out by means of a screw. Two 50-ton steam cranes by Messrs. Cowans, Sheldon & Co., are also situated at this part of the yard. At the steam basin there are a 40-ton steam crane by Messrs. Fairbairn & Co., and a pair of 40-ton sheers worked by steam, the legs of the latter being worked in and out by guys moved by steam crabs; also a 15-ton steam crane and two hand cranes. On the harbor side, there are a pair of 50-ton sheers with fixed legs; and opposite to these, on the side of the ship basin, another pair of 25 tons. In addition to the above, there are several smaller steam and hand cranes and capstans, spread about the yard; also various travelers in the shops worked by ropes, by shafting and by hand.

The travelers in the iron foundry offer a good example of the saving of labor due to the application of power. These 24-ton travelers were originally manual, and required eight men to work them during the pouring of the metal for a moderately large casting. Since power has been applied to them, all the

operations can now be performed by two men at the most, and with much greater certainty.

Comparative Advantages.—Steam power, as already stated, is almost entirely confined to single machines and central stations. If it could be used without being condensed or cooled in the pipes, it would, of course, be more economical than either air or water, because it would save all the efficiency lost in the air compressor or in the hydraulic pump. But it is quite out of the question for any extended general service over such a distance as a couple of miles of pipes, and cannot therefore be considered in comparison with the two other systems in regard to general convenience and economy. Comparing, however, the various advantages possessed by the hydraulic and pneumatic systems, the dockyard experience is greatly in favor of the latter. For a number of machines working with a certain amount of regularity, perhaps the hydraulic system would be the better as regards economy; but with work like that in the dockyard, which includes the use of a large number of machines spread about at considerable distances from the source of power, and with extensive variations in the demands made on the power, it is found that the requirements are best and most economically met by the compressed-air system. By means of the air reservoirs, sufficient power can be stored up to work about half the machines connected therewith for about two hours; the air-compressing engines can therefore be worked at regular speeds, and stopped if necessary for repairs, without fear of the various cranes and capstans being suddenly brought to a standstill for want of pressure; and it is hardly ever necessary for the pumps to be driven at their utmost speed, even when considerably more than ordinary demands are made upon the pressure. Consequently the necessity of working the boiler fires harder for only a few minutes' spurt is avoided, together with the waste of fuel due to steam blowing off at intervals. By pumping the reservoirs up in the evening before the yard closes, power is available for opening a caisson or penstock or working a capstan during the night, without having to start the compressing engines at all. Experiments made here show that the relative

economical values of the air and hydraulic systems, when applied to capstans, are as 13 to 11, respectively, while the great balance of convenience is also in favor of the former; in addition to which it is found that the wear is much less in the air machinery than in the hydraulic, and, moreover, that the engines worked by air are less easily disarranged or put out of working order than those worked by water.

CONVERTED MARINE ENGINES.

[From "Engineering," October 7, 1892.]

An important and interesting change has just been completed in the engines of the steamships *Pallion* and *Stranton*. These vessels were built in 1879 and 1880 for the West Hartlepool Steam Navigation Company, and were fitted with engines by Messrs. T. Richardson & Sons, of Hartlepool, the sizes of the cylinders being 33 inches and 61 inches, with a stroke of 2 feet 9 inches. The working pressure was 75 pounds.

These engines have recently been converted to triple expansion by the same firm, the method of alteration being as follows:

The original boilers were removed and replaced by two of the single-ended type, 12 feet 9 inches in diameter and 9 feet 9 inches long, the working pressure being 170 pounds. Each boiler was fitted with two of Morison's suspension furnaces of 3 feet 9 inches external diameter, and there is no doubt that the high evaporative efficiency obtained from the boilers is due to the furnaces being of large diameter and of minimum thickness, with a comparatively short grate bar. This arrangement allows of more perfect combustion than is possible in a small furnace, and at the same time it offers a length of grate which is easily worked by an average fireman.

The old engines were utilized as far as possible, the original cylinders being retained and used as the intermediate and low-pressure cylinders, the latter being reduced in diameter to 56 inches by means of an independent liner. The condenser, sole plate and pumps were also retained, with the exception of the feed pumps, which were replaced by others suitable for the increased pressure.

A new high-pressure engine complete was fitted to the forward end of the crank shaft, which was altered and provided with an additional crank, the three cranks being set at angles of 120 degrees. The original high-pressure column was removed and replaced by one binding the new high-pressure cylinder to the intermediate, and a casting was bolted to the condenser at the back of the engines, and forms the back support of the high-pressure cylinder. The reversing shaft was lengthened to operate the high-pressure valve gear, and a new reversing engine of the "all-round" type was provided.

The chief feature of the engine, however, is the novel arrangement of the high-pressure cylinder, the improvement being based upon the fact that the greater the circulation over a heat-giving surface the greater is the amount of heat transmitted. The cylinder is jacketed with steam at boiler pressure and the outside of the jacket is surrounded by the first receiver, the design in this respect being similar to the practice in the early days of compound engines, in which the exhaust steam from the one cylinder flowed directly to the steam inlet of the next. The defect in this arrangement was that only a small portion of the steam came into actual contact with the jacket, as the circulation was practically confined to the steam in the direct line of passage from the exhaust port of the high-pressure to the steam port of the low-pressure cylinder. In this engine, however, the receiver is provided with a number of circulating channels which cause the steam to flow uniformly over the whole heat-giving surface of the jacket on its passage to the intermediate pressure engine. The channels are formed by vertical partitions, and the direction of flow being consequently of a zig-zag nature, the steam is continually mixed up, and not only does it abstract a large amount of heat by means of its rapid flow, but the whole body of steam being continually intermingled, it is thoroughly dried before entering the intermediate-pressure cylinder.

It is well known that in the ordinary design of triple-expansion engines a large amount of condensation takes place in the high-pressure cylinder, and that a great deal of water enters the intermediate-pressure cylinder. This is evidenced by the amount

of leakage past the piston-rod and valve spindle glands, and by the rapid destruction of fibrous packing. With the circulating receiver, however, the steam is so thoroughly dried before entering the intermediate-pressure engine that there is no water carried into that cylinder. In the case of these vessels registering drain traps were fitted, but practically no water was drained off.

Another feature in the arrangement is the combination with a steam jacket of an evaporator for producing fresh auxiliary feed water. In this case a Morison's evaporator is connected to the high-pressure jacket at its lowest part, thus making the jacket drain directly into the heating coils of the evaporator, and, in addition to automatically draining the jacket, increasing the circulation of steam therein by the amount used by the evaporator, which steam is still further utilized to heat the feed water on its passage from the hotwell to the feed pumps.

The value of these arrangements, which have been designed by Mr. D. B. Morison, the manager of Messrs. T. Richardson & Sons, is shown by the results attained. In order to obtain reliable data from ordinary working at sea, 140 tons of average coal was put into one of the bunkers at Blyth. All the coal used on the voyage to Port Said was taken from this bunker, and on arrival it was found that the average consumption per day had been 8¼ tons with an average speed of 8.6 knots. Formerly the consumption of coal was 12½ to 13 tons per day, with a speed of 8½ knots.

The reduction of coal consumption was, therefore, 4½ tons per day, or, in other words, for the same speed the coal consumption with compound engines was fully 50 per cent. in excess of the present consumption with the tripled engines. In dull times such as these, the fact that such results can be obtained by tripling the machinery of many of the older ships, should cause shipowners to give the matter their careful consideration. There seems no doubt that many ships now working at a great disadvantage with compound engines might be made to show a satisfactory dividend if the machinery was converted as successfully as in the cases we have related.

THE LARGEST DRY DOCK ON THE LAKES.

[From "Inland Ocean."]

The great dry dock of the American Steel Barge Co. has been completed, and water was let into it for the first time early in October. Superior can now boast of having the largest dry dock on fresh water, the measurement being 500 feet long on the keel blocks, 90 feet wide and 18 feet deep on blocks. It is 100 feet longer than any other other dry dock on the Great Lakes, and will hold at one time two of the largest vessels on the lakes.

It will be a great convenience to vesselmen, as they have heretofore been compelled to go to Detroit and Buffalo to make any repairs. A channel has been dredged to the dock, and now everything is in readiness for work.

The engine house stands at the entrance, and is supplied with two Porter engines of 125 horse power, which have a pumping capacity of 16,000 gallons a minute. They can empty the dock in two and a half hours.

REFRIGERATING MACHINERY.

One of the latest improvements in machinery of this kind is the use of carbonic anhydride, which appears likely to supersede the system in which compressed air is used as the working medium.

This system was introduced in England by Messrs. J. & E. Hall, of Dartford. It was first brought out in Germany, where its success led this firm to develop it in England. The principle upon which the machine works is the evaporation of carbonic anhydride, which is constantly recompressed and liquefied for further evaporation. The compressed gas is delivered into a condenser, consisting of coils of tubes kept cool by circulating water. The evaporation is produced by allowing the liquid carbonic anhydride to enter coils which are continually being exhausted by the compressor pump. Thus, the pressure being insufficient to maintain the gas in a liquid state, it evaporates at a low temperature, cooling the brine surrounding these coils. The brine, thus reduced to any desirable low temperature, is circulated by means of pumps through pipes in cold storage cham-

bers, ice-making tanks, or is used for any other refrigerating purpose, the brine absorbing the heat from the surrounding objects.

Liquid carbonic acid is now obtainable at so very low a price that it commends itself for this purpose, and the refrigerating effected by its use costs for fuel only about one-fifth the cost of fuel for refrigerating by compressed air.

J. & E. Hall's patent carbonic anhydride refrigerating machines are made in pairs and fitted with compound steam cylinders arranged side by side, having a surface steam condenser between them. The compressors are placed in line with the steam cylinders, the cranks of which are at right angles, by which arrangement an even turning moment is obtained. Each compressor delivers its gas to an independent condenser, contained in the base of the machine, in which the carbonic anhydride is condensed in the liquid form. In connection with each condenser there is a separate refrigerator, consisting of coils of wrought-iron pipes, in which the liquid carbonic anhydride evaporates, producing intense cold. These coils are contained in a steel casing in which the brine is circulated.

Carbonic anhydride, unlike ammonia, has no affinity for copper, hence that material can be used to resist the action of the sea water, which rapidly corrodes wrought iron, a point of the greatest importance when valuable cargoes are entrusted to the machine.

The machine above mentioned, when running at its normal speed of 80 revolutions per minute, indicates 70 H.P. The suction and delivery valves are made of tool steel, the valve being coned at 45°, whilst the seats, also of steel, are rounded, so that the bearing surface is very narrow. In this machine, even the neglect of the attendant appears to be provided for. Thus, in the event of any other machine being started without the delivery screw-down valve being previously opened, a breakdown having serious results might occur; but in this machine a safety valve has been fitted, preventing all possibility of accident.

The joints between the various parts of the refrigerating machine are absolutely tight, being made of any material suitable

for the purpose, according to the working conditions. This is a feature with the carbonic anhydride machines, carbonic acid gas attacking no metal or material, whereas with ammonia and similar refrigerating agents, only iron or steel parts can be used.

The various parts of the machine are very much smaller than is the case with ammonia machines, whereby the strength is very greatly increased.

In these machines the portions are so arranged in duplicate that either the entire plant can be worked, or, if desired, one of the compressors, with its condenser and refrigerator, can be disconnected, when the other compressor can be worked by itself, the full advantage and economy of the compound engines being retained. Another point of advantage is that this duplex system gives all the security of two independent machines.

This type of machine has been designed to meet the limited space available on board ship, and to suit the usual height of the 'tween decks, machines even of the largest type, capable of preserving up to 60,000 frozen sheep, being less than 6 feet 6 inches in height.

It may be interesting to note a few facts with regard to carbonic anhydride. Twenty years ago the production of this liquid was a laboratory experiment; it is now sold at a few cents per pound, and large companies are formed in many parts of the world for its manufacture and sale. It is inodorous and non-poisonous in the quantity used. To give an idea of the freedom from danger of this class of machine, it may be stated that the entire contents of the machine might be allowed to escape into an ordinary engine room without any of the disastrous results which would follow a similar escape of ammonia, sulphurous acid, ether, &c., used in other machines.

Of course a small quantity is required to be added to the charge from time to time to replace losses, but this is most easily done even in the remote parts of the world, as the carbonic anhydride is sent to any part of the world, in which it cannot already be obtained, in steel cylinders. To give an idea of the cost of first charging a 24-ton ice plant with the gas, it is understood that it does not exceed, in England, $40.

The success of these machines may be judged from the fact that although it is only three years since Messrs. Hall placed the first machine on the market, there are now some 65 machines being used by the largest meat-freezing works and importers in the world.

THE BRITISH NAVAL MANŒUVRES, 1892.

[Extract from "Marine Engineer."]

Lastly, as regards the casualties. We have lost a torpedo boat by collision. But, with so many boats engaged, that is scarcely to be wondered at. Our neighbors have had more numerous collisions, though, thanks to the fact that they did not occur in deep water, the boats have been raised. The boiler of another boat gave way, and three men were injured. There was an accident, too, to the machinery of the *Arethusa*. Possibly, when further details of these accidents are to hand, we may learn a lesson from their occurrence. The battle ships behaved well, and were free from casualties. But the cruisers and torpedo gun vessels were the chief sufferers. Foremost amongst these was the torpedo catcher *Spanker*, a vessel rated at 20 knots, only two years old, and incapable of attaining a greater speed at sea than 12 knots. She is always breaking down. Not far behind her in the race for the last place comes the similar vessel *Sharpshooter*, which had a narrow escape coming home, and got 70 tons of water on board, mostly by leakage. The manœuvres have shown, if they have done no other good, that these vessels—rightly might they be called the *Gossamer* class, after one of their number—are too flimsy and light for their work.

The accidents to the cruisers amount to very little. Like the collisions between torpedo boats, runnings ashore must happen under war conditions, when vessels have to go full speed in fogs off dangerous coasts. If the war conditions are not observed, and the ordinary precautions of slow speed are taken by cruisers sent on important missions, the manœuvres lose the greater part of their value. We must conclude, on the whole, that the events of the month show that the ships are, as a body, in a satisfactory condition, and that our officers and men are as smart and seamanlike as in the brave days of old.

THE SCANTLINGS OF SHIPS' BULKHEADS

[" Engineering," October 14, 1892.]

One of the most important questions which engages the attention of naval architects is the rendering of ships unsinkable by provision against localized fractures in the shell plating of the hull, and although bulkheads are universally regarded as meeting the case, the method of construction, or rather of stiffening these, gives rise to difference of opinion. This was abundantly manifested in the report of the technical members of the Bulkhead Committee. Any information, or, preferably, tests bearing on the subject, must be of interest. Flanged plates, instead of plates stiffened by angles, were being used, and Lloyd's, with the anxious care to stimulate improvement which distinguishes them, resolved to test the new plan against the old system. Messrs. A. and J. Inglis, the well-known shipbuilders in Glasgow, having the newest and most improved flanger constructed by Messrs. Hugh Smith and Son, Possil Works, Glasgow, were requested to carry out the experiments at Lloyd's expense and under the inspection of Lloyd's surveyors. There were constructed two strips of bulkheads with plates similar in thickness; but while the one was stiffened by an angle, the other depended entirely for stiffening on the flange. The strips were rigidly connected at top and bottom, and were forced apart by a screw jack, placed in the middle of the length of the plates and against the angle and flange, respectively. The angle-stiffened bulkhead deflected sixteen times as much as the flanged one, a striking difference. In oil ships, where it is so important to maintain the caulking, the advantage of this rigidity is so great that it is certain to be adopted where practicable in future tank vessels.

RATIONAL DESIGN OF COMBUSTION CHAMBERS AND TUBE SHEETS FOR FORCED-DRAFT BOILERS.

[A letter to " Engineering," published October 21, 1892.]

To the Editor of " Engineering."

Sir: The application of forced draft to the furnaces of marine boilers has certainly not proved so advantageous as was at first anticipated, owing to the increased temperature in the combustion

chamber causing the tubes to leak, but there can be no question as to its great advantage, even to the limited extent of an air pressure equal to a column of ½ inch of water, which is now termed "natural draft," for without the means of supplying what may be more correctly termed "mechanical draft" to the furnaces, the evaporative capability of a boiler is subject to considerable fluctuation, owing to the fires and tubes becoming dirty when steaming, to different qualities of coal, or to the absence of sufficient down draft to the stokeholds to supply the necessary amount of air for proper combustion.

When forced draft was first adopted boilers were designed to withstand an air pressure in the stokehold equal to 2 inches in the water gauge, by which means the quantity of coal burnt per square foot of grate surface could be more than doubled, and as this excessive consumption would, in ships of war, be only occasionally required, it was decided that, in order to lessen the weights carried on board and economize valuable space, the capacity and weight of the boilers could be reduced.

I have heard many captains and other naval officers question the advantage of forced draft, which is, in my opinion, only natural, considering the disastrous effects often arising therefrom; and they have suggested larger and heavier boilers as the only means of obtaining high speeds with safety. This, if adopted, would be descending the ladder we have set ourselves to climb; and, although difficulties have arisen owing to want of experience, I am quite certain they will be overcome, and I trust this country will not be second in the race.

Now, in dealing with this apparently difficult question, the first thing to be considered is the cause of the tubes leaking. Several reasons have been advanced, such as bad stoking, cold air being admitted to the furnaces, or the deposit of fatty matter on the tube plate. All of these should be, as far as possible, avoided, but they have been abundantly proved not to be the real cause.

The Admiralty have recently adopted a ferrule, which has, to a certain extent, been successful; but it can, at best, only be looked upon as a makeshift, for we cannot allow that our engine

designers have already thrown up the sponge by admitting their inability to design an efficient boiler, and it is in the direction of design that we must look for improvement, by considering wherein the present defects exist.

In my opinion, they are to be found in the construction of combustion chambers of all boilers of various types; but, before enumerating the improvements I would suggest, I will first consider what, in all probability, takes place in the chamber when heat is applied, taking the double-ended boiler, with common combustion chamber, as being the one in which defect is most apparent.

The tube plates are subjected to various strains, all tending to curve them outwards, toward the fire, viz.:

1. The difference of temperature between the inner and outer surfaces.

2. The lengthening of the tubes due to expansion.

3. The expansion of the top and sides of the combustion chamber, tending to force back the outer edge of the tube plates.

4. The internal steam pressure.

In boilers as now designed the first three of these forces are ignored, and the fourth alone is provided against by fitting stay-tubes at intervals. This, in my opinion, causes the tube plate to buckle when heated, the plate, in all probability, taking a different form each time steam is raised, or the temperature of the combustion chamber suddenly increased or diminished, and this may account for the tubes often giving out after two or three hours on a forced draft trial. I have observed that the defect of leaky tubes is more prevalent when boilers are first tried, and that often, after they have been used a few times, the difficulty diminishes, provided they are worked with judgment and not subjected to any great increase of air pressure. This I attribute to the tubes being fixed in place when the tube plate is cold—the plate altering its form when heated, loosening the tubes in the holes; after the boiler has been in use a short time, and the leaky tubes re-rolled, the tube plate in all probability assumes a more definite shape, and, when heated, alters in a similar manner each time, provided the conditions are the same, but with air

pressures varying from ½ inch to 2 inches this cannot be depended upon.

Alterations of form, due to expansion, took place, in a minor degree, before the introduction of forced draft, when leaky tubes were practically unknown—but it must be remembered we then had lower temperatures and air pressures to contend against, and brass tubes to help us—and the latter were of great advantage, being sufficiently ductile to accommodate themselves to the altering form of the tube plate. Even now, if the duration of the boilers were not considered, the adoption of brass tubes would probably overcome, to a great extent, the present difficulty—in support of which I am able to state that I had two first-class torpedo boats under my supervision, running almost daily for over two years, for instructional purposes, one fitted with brass, the other with steel tubes. Such a thing as a leaky tube was unknown in the former, while in the latter the tubes were always leaking and had to be renewed, with no better results. But the use of brass tubes is not recommended, as it is practically impossible to avoid the galvanic action between the two metals, resulting in a rapid deterioration of the boiler stays and shell; therefore, to avoid this, all the material used in the construction of a boiler should be as nearly as possible of the same composition, and as mild steel has been decided upon as the most suitable, the tubes should also be of that metal, and, I think, with a slight modification in the present designs, they would meet all requirements.

In the first place, the expansion of the combustion chamber should be provided for by putting a corrugation either round the outer flanged edge of the tube plate, extending below the lower row of tubes, or in the side plate of the chamber itself; secondly, the thickness of the tube plate should not exceed ½ inch; and lastly, most important of all is the method of fixing the tubes at the combustion chamber end. The old rough-and-ready method of simply expanding the tube ends in place must be discarded, as being inadequate to meet the increased strain consequent upon the use of forced draft, and special tubes as stays be done away with, as they only tend to distort the tube plate, causing the

others to leak. Every tube ought to be of equal thickness and form a stay of itself, by being properly fitted and secured in place as follows: When the holes in the tube plate are bored they should be faced on the inner side to form a seating of about $\frac{3}{32}$ inch wide, and a groove formed around the hole on the outer side of the plate. The ends of the tubes should be upset to admit of a shoulder being turned on them to bear against the seating, then lightly rolled in place, and the ends riveted over and beaded into the groove. This would secure a perfect metallic joint, and the connecting surface between the tube and tube plate would be considerably increased, which is a most important consideration for the free and rapid transfer of heat from the one to the other, and it would prevent the ends of the tubes being burnt away. The other ends of the tubes could be fixed either by expanding and beading in the usual way or by screwing, the holes in the plate being made large enough to allow the shoulders on the other ends of the tubes to pass through.

I feel confident that if this system were adopted no more mishaps through leaky tubes would occur, and it would admit of a much higher air pressure being used; the weight and capacity of the boilers could also then be reduced without impairing their efficiency.

It must be apparent to every engineer that the adoption of higher pressures and temperatures calls for greater skill and intelligence on the part of the workmen employed in the manufacture of our boilers, and the method of fixing the tubes proposed above would require special attention. All the tubes should be firmly fixed at the combustion chamber end before any attempt is made to secure the other ends—unless the method of screwing were adopted, which I do not consider at all necessary. As, in my opinion, the tube plate is the chief cause of the present troubles, it ought to have our first consideration, instead of which it is subjected to every possible ill-treatment. It is stayed at intervals by stay tubes, causing it to buckle, and it is rigidly stayed on each side to the boiler shell, while the upper part is left free to expand upwards, tending to elongate the holes in that direction. Now, by having corrugations, as suggested above, these disad-

vantages would be avoided, and the expansion of the chamber side plates would be independent of the tubes and tube plate.

I also contend that large flat-top combustion chambers are most undesirable, especially where great heat is applied; for when the water in the boiler becomes low, say, only covering the top of the chamber with a depth of from 3 inches to 4 inches, and the ship rolling, a large portion of the upper surface becomes uncovered and therefore unduly heated, which would not be the case if it were curved.

Also the number of stays required to support it materially interferes with the proper circulation of the water, and I feel confident that much of the bulging which has taken place in the boilers of vessels of the *Spanker* class may be attributed to this method of construction.

H. BENBOW,
Chief Inspector of Machinery, Royal Navy.

THE ATLANTIC RECORD.—THE RECORD FROM QUEENSTOWN AGAIN BROKEN.

NEW YORK, Oct. 19.—The steamship *City of Paris*, of the Inman line, has again broken the westward record from Queenstown, which was made by her when she made the run in July in five days, fifteen hours and fifty-eight minutes. She also beats the record for single day's run, by covering 530 miles against the record of 528 miles made by the *Teutonic*. The *City of Paris* sailed from Liverpool at 1.30 P. M. on October 12, and from Queenstown October 13, and arrived at Sandy Hook lightship a little after midnight last night, covering the distance of 2,782 miles in five days, fourteen hours and twenty-four minutes, beating the record by one hour and thirty-four minutes. Her days' runs were as follows: 448, 508, 503, 505, 530 and 288. Average speed per hour for the whole trip, 20.7 knots; average speed per hour for best day's run, (allowing for .785 hour gain in time), 21.385 knots.

In connection with this performance, as well as those of the *Paris* and *Teutonic* given in the August number of the JOURNAL,

(pp. 494–6), the annexed editorial from the "American Machinist," of September 1st, 1892, is very interesting:

THE *TEUTONIC'S* 528-MILE RECORD.

"In our issue of August 11th we referred to the new record of the *City of Paris*, and to that of the *Teutonic* for a single day's run, this latter record seeming upon the face of it to prove her a faster vessel than the *Columbia*, which is being built for the especial purpose of overhauling any merchant vessel. It has since been pointed out, however, that the *Teutonic's* record was made in such a high latitude, and moving in such a direction, as to make the actual time from noon to noon, as determined by meridional observation, 24.93 hours instead of 24 hours, and that the actual speed of the vessel was, therefore, somewhat less than the contract speed of the *Columbia*, instead of greater, even on the supposition that there was no mistake of any kind.

"It will be remembered that last year the *Teutonic* claimed to have crossed in time which was disputed, some doubt being thrown upon the claims of the ship's officers by those stationed upon land, both as to time of departure from the other side and arrival at this. The incident is brought to mind by a letter received by us from the well-known engineer and manufacturer, Mr. Geo. B. Grant, of Lexington, Mass. Mr. Grant says:

"'Referring to your editorial concerning the record of 528 miles run in one day by the *Teutonic*, I beg leave to express certain doubts based on observation. I was a passenger on the ship at the time.

"'We started out from Queenstown with a rush, steam blowing off every few minutes, ship shaking and wiggling with the 'screw motion,' great breakers to the right and left of the bows, screw making eighty-three to eighty-five revolutions, and everything right up to high pressure. The *City of Berlin* was far in advance, and presently she was far in the rear. The ladies looked anxious, and the pools ran on the high numbers. Mind, the pools ran high, for there's the point.

"'The pool is no part of the engine, but if I mistake not it has something to do with the record. For the benefit of honest peo-

ple I will explain the pool, and show its operation as well as I can from observation, not participation. There is a gambling saloon on board the ship. It is behind the rum shop, and is called the 'smoking room.' There are little tables fitted up with pots and partitions that are so convenient for smoking, and when the gamblers are tired of smoking poker they have a pool for a change between drinks.

"'The pool is a betting scheme on the ship's log for the next day. Some twenty-five numbers are sold at auction, one by one, each to the highest bidder, and the total sum received forms the pool. If you want to bet that the ship will run 503 miles during the current day, you bid high enough to buy the number 503. The log is posted the next day at noon, and if it reads 503 miles you take the whole pool and have successfully transferred many pounds sterling from other pockets to your own. If on Sunday, you are expected to show a proper respect for the day by not betting, at least not in the gambling saloon.

"'Well, we started off with a rush and had splendid weather for about three days. Every one knew that we were after that record for all we were worth, and that the ship was able to take it if circumstances would so combine as to give her a fair chance. Every one was interested and noted the tremendous exertion that was being made by the eighteen thousand horse-power engines. The pools sold high.

"'The log was a low one in spite of the evident hurry and extra favorable conditions. The lowest figure took the pool. Three good days and three low records. Our chances for beating the *City of Paris* began to look slim, and pools sold low. We had a quiet run over the Banks in the fog, and a low record. Next day we had cleared the Banks and were running quietly in a fog that was almost continuous. Steam was not blowing off, but the fog whistle was about all the time. The screw motion was much less than at first. The screw was running about eighty revolutions. During the night the speed was slackened at times, so that it was not evident to the passenger in his berth that the ship was going at all. We all expected a low record for the day, but, nevertheless, it was 528 miles.

"'Now, I did not take the latitude and longitude at noon every day, and therefore I do not know but that the ship made 528 miles that day. I do not say that she did make that record. But I do say that the exertion during that day was not as apparent to the observer; that the breakers at the bows of the ship were not as great; that the speed of the screw was not as great; and that the foggy condition of the weather was not as favorable as on the first three days' run.

"'I also wish to express my doubts of the honesty of anything and everything that has any connection with a gambling saloon. I do not say that the ship's log was cut down a little for the first three days, and the shavings added to the last day, but I do say that the parties interested in the gambling operations would manipulate that log in just that way if by any means they could get the chance. I do not know whether the officer having the log in charge was interested in the gambling or not, but if he was so interested he could easily earn more than his salary by doctoring the record, and could do it in such a way that no one could prove him wrong.

"'Now, consider. The ship was after the record. The record is worth a great sum to the holder. We should have got the desired figure if the engine had not broken down the evening before reaching New York. Is it reasonable that the ship would have run slow for three days and then speeded up for the last day? Was it not reasonable that the ship would make as many miles as possible at the start, to be sure of them against the chances of fog on the Banks? If the gamblers had any chance to do it, would they not work the daily log so as to fleece the lambs in the smoking room, and yet come out square with the record for the whole run? Do professional gamblers ever lose a chance to cheat? Did they have a chance in that case? (If the ship could not capture the desired record for the whole trip, was it not desirable to get it for at least one day, on the principle that half a loaf is better than none? Possibly the *Columbia* will be able to overhaul the *Teutonic*, after all.)

"'Excuse me, but I have my doubts as to that day's run.'

"We think the above letter is a very significant one. We do

not know that Mr. Grant has had much, if any, experience in navigating ships, and probably he has had none, but he has the engineer's habit of inquiring into the reason of things and connecting cause and effect. His letter is well worth thinking about, especially by those who are responsible for the things he mentions. At first, the pools upon the day's runs of vessels included small amounts only, and served to pass the time away for many people who, being devoid of mental resources and idle-brained, must always have something going on to "kill time." But we all know that gambling operations seldom or never stop at this stage. The fever gradually rises, and the amounts staked increase until the professional gambler is attracted, and the professional gambler always operates in some such way as indicated by Mr. Grant.

"It is important to the engineering world to know just what a given engine or a given steamship is capable of, and it is very much to be regretted if the professional gambler is to be allowed to becloud the subject. But, on the other hand, it would be well enough for the p. g. to know, perhaps, that in meddling with something in which engineers are concerned he will in all probability have some new experiences. It is a part of an engineer's business to analyze records of performances of machinery."

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*OUR NEW CRUISERS AND THEIR COAL SUPPLY.

Although much has been accomplished during recent years towards economizing fuel by the introduction of improved machinery, yet the proportion of coal carried at the present time in our warships to power developed is much less than formerly. In point of fact, the amount of coal carried by the twenty-four new second-class cruisers is considerably less than is necessary for them to keep the sea, or even to make a voyage of a little over 2,000 miles at a not excessively high rate of speed. This constitutes a lamentable blemish which, now that these vessels are either fitting out or ready for service, admits of no cure. To

*British Navy.

verify this statement it is only necessary to examine the report of the *Latona's* performances during an experimental voyage to Malta and back with the late Controller of the Navy, Vice-Admiral Hopkins, and Rear-Admiral Morant on board. This return clearly shows that the coal supply of these cruisers is inadequate for the work they are required to perform; and it is made by such a high authority that even admiralty officials must accept it as conclusive.

It must not be forgotten that all this class were designed to steam 20 knots with 9,000 I.H.P., and 18.5 knots with 7,000 I.H.P., and be capable of maintaining a continuous sea-going speed of 16 knots with 3,750 I.H.P. Notwithstanding this enormous power available, the coal capacity provided for at the load-line is only 400 tons—an insufficient quantity to steam from Plymouth to Malta at a speed of 16 knots. This is somewhat startling, but the following facts, officially stated, confirm this statement to the letter.

From Plymouth to the Rock measures 1,050 miles. The *Latona* averaged 15 knots during the passage. The expenditure of coal was no less than 263 tons, or about 90 tons a day, although the machinery was working up to 3,200 I.H.P. only for the whole of the passage, with the exception of the last 18½ hours before reaching Gibraltar, when 3,900 was developed. At the Rock 260 tons of fuel were taken in, and then the run to Malta began. From a steady 14½ knots the speed was increased, and a twenty-four hours' spin was made, during which period 401 miles were covered, or an average of 16.6 knots, with 4,300 I.H.P. This is very satisfactory, but the value of the report would be considerably enhanced if the consumption of coal were also given for this day's run. At all events, 200 tons more coal were required and shipped at Malta. Thus, after steaming under 2,000 miles at an average speed of just over 15 knots, 463 tons of coal were burnt —just 63 tons more than the coal capacity at load draught, as laid down in the Navy estimates. Proceeding back to Gibraltar at the more leisurely speed of 11.3 knots, and taking 87.5 hours over the passage, 190 tons more were received on board—a total of 653 tons for a run of something under 3,000 miles.

Again dismissing as improbable that these cruisers will ever again work up to 9,000 horse-power under the destructive agency of forced draft, still, their radius of action at 7,000 I.H.P., on natural draft, may be calculated with a very fair amount of accuracy. Taking a consumption of 2 pounds of coal per I.H.P. per hour, and allowing the by no means extravagant estimate of five tons a day for electric light, distilling, cooking, &c., we have a total expenditure of 155 tons in 24 hours. Supposing these cruisers' *actual* stowage is 550 tons, then this supply only means a barely sufficient quantity for 3.6 days' steaming. Taking the corresponding speed—in smooth water—at 18.5 knots, or 444 miles a day, this cruiser's coal supply would give out after steaming 1,598 miles!

Nor is there any romancing in the information derived from an official report. It is always exceedingly difficult to obtain any accurate knowledge of the coal consumption of our newer ships, especially at the higher speeds. In the case of the *Latona*, her failure as a stayer is manifest. This exhibition of defectiveness in a design, commendable in all other respects, is deserving of the most serious attention. Sir Edward Reed, than whom no one has striven more to obtain a larger coal supply than at present carried in our war ships, could hardly be exposed to a greater shock than the one imparted by the disclosure of these facts. Nor is this all. As an additional proof of the manner in which our cruisers are being stinted in their coal supply, the improved cruisers of the *Astrea* class, some 960 tons larger than the *Apollos*, are only credited with the same coal capacity at the load draught in the current year's estimates. Would it not be a worthy sacrifice to eliminate tons of comparatively useless lumber which now so seriously hamper our ships? The room thus gained, if filled with fuel, would conduce to a more satisfactory and extended mileage performance than the one now under notice.—"The Naval and Military Record," Sep. 22, 1892.

SAIL POWER FOR *OUR NEW NAVY.

The controversy as to the use of sail power in modern naval vessels does not appear to be finally settled, in spite of the fact that the policy of reducing it to the minimum has been adopted for our new Navy. An argument for an increase of sail power has been founded upon a statement credited to an officer of the *Dolphin* that she could not have crossed the Pacific had she not had some sail power, as she reached Yokohama with but sixteen tons of coal in her bunkers. The answer to this by the opponents of sails is that the bunker capacity should be increased; that even the sixteen tons of coal she saved was good for two hundred knots at a speed of eight knots, or the equivalent of four days' sailing. On one occasion the *Dolphin* used sails, while adjusting the bearings of her engines, en route from San Francisco to Honolulu, and though she had the full force of the northeast trades in her favor, she could barely keep steerage way, making but two knots an hour. With the engines running at eight knots, sails would not add more than half a knot to the speed of the vessel.

Perhaps we are trying to get too much out of our small ships, and probably the space for coal might be profitably increased at the expense of some other features. It is evident that the coal consumption at low speeds has been underestimated, and in the new ships allowance will be made for this and the bunker capacity enlarged. The most economical peace cruiser will, no doubt, be found to be a vessel of about two thousand tons, having a maximum speed of thirteen or fourteen knots. Machinery for this power at forced draft would be comparatively small, and when worked for a speed of say ten to eleven knots would be at nearly full power natural draft. As the boilers and engines would be designed for this particular power, they would work with much greater economy than higher powered machinery at the same lower power. The space and weight saved in the machinery could be put in coal, so that we should have a greater supply at the same time that we were spending

*United States Navy.

it more ecomically. Such a vessel would be of the greatest service in peace, but, of course, in time of war, she would be so slow as to be an easy prey for a fast cruiser, and would have to be laid up at home. It would seem, however, that there is room for a considerable number of such ships, and that in case of a long peace, they would more than pay for themselves, even if, in time of war, they have to be laid up. Another point against sails is that even when the wind is favorable, it may be very light. But when it is dead ahead, as is very often the case, then the spars and rigging add a very decided amount to the resistance. One of our ships which had been some time out of dock, and which was fitted with about three-fourths of full sail power, owing to her foul bottom had to send down the lower and topsail yards to enable her to proceed against a fairly strong head wind. Attention is directed in England, too, to the inadequate coal capacity of the twenty-four new second-class cruisers, and it is urged that it would be well to eliminate some tons of comparatively useless timber and substitute additional coal bunkers. As it is, these vessels are unable to keep the sea for any length of time, or even to make a long voyage.—"Army and Navy Journal," Oct. 15, 1892.

PRESERVATION OF BOILERS WHEN NOT IN USE.

Under date of April 14, 1892, an order was issued for the preservation of boilers of the French Navy when not in use, of which an abstract is given below:

On board all the ships in the reserve, as well as on those which are laid up, the boilers will be completely filled with fresh water, and this is to apply to shell boilers as well as those of the tubulous or pipe type. In the case of large boilers with large tubes, there will be added to the water a certain amount of milk of lime, following the instructions furnished by Belleville & Co. for the preservation of the tubes of their boilers, or a solution of soda may be used instead. This treatment will also be followed for boilers in the storehouses on shore.

In the case of tubulous boilers with small tubes, milk of lime

or soda will be added, but the solution will not be so strong as in the case of the larger tubes, so as to avoid any danger of contracting the effective area by a deposit from the solution. The strength of the solution will be just sufficient to neutralize any acidity of the water.

Care must also be taken to preserve the outside of the steel or iron tubes in the case of boilers which are not to be used for long periods. To this end, they will be painted with red lead or coal tar, as far as it is possible to reach. For those places which are inaccessible, a protective coating will be obtained by burning, under the tubes, a certain amount of tar or coal tar, whose dense smoke will form a coating of soot, and thus prevent the air from reaching the surface of the tubes. In addition, the boiler casing will be closed and kept airtight after placing inside some quicklime.

Periodical inspections of boilers thus treated will be made to assure the complete filling of the tubes with water.

THE LARGE STEAMSHIP COMPANIES OF THE WORLD.

["New York Times."]

The new list of the merchant marine compiled by the Bureau Veritas, which is now in press, gives some interesting statistics concerning the great ocean steamship companies of the world.

Some people will be surprised to learn that the transatlantic lines are not the leading ones in aggregate tonnage or number of vessels. The British India, of Glasgow, and the British India Steam Navigation Company, and the British India Associated Steamers, all one concern, head the list with 103 steamships. Their aggregate measurement is 239,635 tons. Next stands the Peninsular and Oriental Steamship Company, with 36 steamships, representing 216,887 tons. Next in order comes the Compagnie des Messageries Maritimes, with 61 vessels, whose aggregate tonnage is 202,522. The fourth on the list is the Norddeutscher Lloyd, of Bremen, some of whose vessels ply to this port. This company has 66 steamers, measuring in all 197,169 tons.

Other important lines are the Navigazione Generale Italiana, 106 steamers, 170,755 tons; the Compagnie Generale Transatlantique, 66 steamers, 167,487 tons; the Hamburg-American Packet Company, 56 steamers, 165,442 tons; the Wilson Line, 86 steamers, 158,089 tons; the Austrian Lloyd, 73 steamers, 128,359 tons; the Allan Line, 39 steamers, 123,038 tons; the Spanish Compania Transatlantica, 34 steamers, 94,163 tons; the White Star, 20 steamers, 95,370 tons, and the Cunard, 26 steamers, 85,913 tons.

LOSS OF HEAT BY SMOKE.

[From "Engineering," October 28, 1892.]

It is often rashly asserted that the production of smoke shows a sensible waste of fuel, but according to the very careful experiments of M. Scheurer-Kestner this is not so. Even when the smoke is purposely made as dense as possible, the waste of combustible material is under 1½ per cent., and under ordinary conditions it is, of course, much less. It is not sufficient to insure the absence of smoke that the air supply to the furnace is properly regulated, as smoke is produced in two different ways. In the first place, rapidly heating the hydrocarbons of the fuel on the grate causes them to break up with deposition of carbon, and further, many hydrocarbons, in meeting the cold surfaces of the boilers, also undergo decomposition with a further deposition of carbon. The best way to reduce the smoke to a minimum is, according to M. Scheurer-Kestner, to adopt a suitable mechanical stoker, or to burn the smoke by providing for its being thoroughly mixed with very hot gases.

SHIP-LIGHTING PLANT.

[From "Marine Engineer."]

An improved pattern of ship-lighting plant has been brought out by Messrs. W. H. Allen & Co., of the York Street Works, Lambeth, S. E., London.

One style is an 8-inch by 6-inch vertical engine, driving direct a continuous current dynamo. The output of the dynamo is 130 ampères at a pressure of 62 volts, and it is therefore capa-

ble of maintaining alight 130 16-c.p. lamps. Similar sets are made in standard sizes for 60, 90, 160 and 200 lamps, and are in every way adapted to the requirements of the larger cargo steamers. The engine is controlled by an automatic expansion governor, which varies the cut-off of the steam distribution valve, and so proportions the steam admitted to the cylinder to the work required; further, this governor embodies a recent patent of Messrs. Allen, by which alteration of the speed can be effected by hand regulation while the engine is running. One of the advantages of this hand regulation is to counteract any change of speed which may occur when the engine ceases to be connected to the main condenser, and exhausts into the atmosphere when in port.

Special attention has been given in the design of these dynamos to the two points which must decide the behavior of a dynamo when running under the somewhat trying conditions which prevail on shipboard. "Without over-heating" and "without perceptible sparking" have now appeared so often in specifications and description as to have lost whatever little meaning they ever had; it is, therefore, well to gain some definite ideas as to what may or may not be obtained. The British Admiralty specify that a run of six hours' duration, at full load, should be made without any accessible part of the dynamo reaching a temperature of more than 70 degrees Fahr above that of the surrounding air of the room, and, further, that this room may have a temperature of 120 degrees Fahr. The rise of temperature in these dynamos falls well within this limit, and they may, therefore, be regarded as thoroughly trustworthy and durable when working in a ship's engine room at any ordinary or even extraordinary temperature. Quite as important as the heating of the armatures, or field-winding, is the question of sparking at the brushes, as affecting the life of the commutator, and so the cost of repairs. In central stations, where dynamo attendants are always at hand, a dynamo which does not show sparking at full load, when the brushes are adjusted to the correct position, is practically all that is required, but on board ship something more should be attained, if possible, and it should be possible to leave

the brushes in an incorrect position without having to continually alter them for every change of load. In the dynamo in question no sparking will ensue if the brushes are not shifted, whatever may be the load between the limits of quarter and full load. This result is obtained by a careful proportioning and shaping of the pole pieces, and, further, by the division of the armature winding into a large number of sections, the number of sections in the commutator being correspondingly great. A single loop of the winding is attached to the commutator section in the drum armature, and in the ring armatures for smaller machines only two turns per commutator section are used.

A larger engine and dynamo, similar to those already described, was supplied to the Orient Steamship Co. for the lighting of the *Ophir*. Each of the four sets on board consists of a compound engine, having cylinders 8 inches, and 13½ inches diameter by 7 inches stroke, driving directly compound-wound dynamos with drum armatures, all of the firm's manufacture. The output of each machine is 300 ampères, at a pressure of 67 volts, requiring about 35 H.P., the over-all efficiency being extremely high, and the engines being very economical in their consumption of steam. The speed is no more than 230 revolutions per minute, and the floor space occupied by the combination is 8 feet 11 inches, by 3 feet. When feeding the lamp circuits, three dynamos are worked in parallel, the fourth being held in reserve. The temperature of the room in which the dynamos are placed is normally 110 degress Fahr., and frequently reaches as much as 130 degrees, yet in this temperature the working of both engines and dynamos gives entire satisfaction, while the parallel working of the dynamos has proved itself of great service in minimizing the amount of attention required.

We are informed that the largest passenger steamer and the largest ship for H. M. Navy, launched last year, were, in both instances, supplied with engines and dynamos manufactured by Messrs. Allen.

SHIPS.

UNITED STATES.

Brooklyn (Armored Cruiser No. 3).—This vessel is in a general way a duplicate of the *New York*, some changes having been made since the latter was designed.

Length on load water line (normal displacement),	400 feet 6 inches.
Moulded breadth at load water line,	64 feet 0 inches.
Mean draught at normal displacement,	24 feet 0 inches.
Freeboard at bow,	29 feet 10 inches.
Freeboard at stern,	21 feet 4 inches.
Normal load displacement, about,	9,150 tons.
Total coal capacity,	1,650 tons.
Coal carried at normal load displacement,	900 tons.
Complement of officers and men,	566.

The speed to be maintained for four consecutive hours, under conditions to be prescribed by the Secretary of the Navy, is not to be less than an average of 20 knots an hour, and for every quarter knot exhibited above that speed the contractors are to receive a premium, over and above the contract price, of fifty thousand dollars ($50,000.00); and for every quarter knot that the vessel fails of reaching the average speed of 20 knots, there will be deducted from the contract price the sum of fifty thousand dollars ($50,000.00). The speed trials are to take place with a displacement of eighty-one hundred and fifty (8,150) tons.

The hull is to be of steel, not sheathed, with double bottom and close water-tight subdivision to about 12 feet above the water line.

There will be two military masts, with fighting tops, to carry no sail.

Protection of the hull is to be afforded by means of a steel protective deck, worked from stem to stern, and supported by heavy beams. The bottom edges of this deck amidships are to be 5 feet 6 inches below the 24-feet water line, the top of the deck rising to this water line at the center of the vessel.

On the slopes of the deck, over machinery and boilers, the armor is to be 6 inches thick; on the horizontal portions, it is to be 3 inches thick; forward and abaft the machinery and boilers, to stem and stern, the deck is to be at the thinnest part at least 2½ inches in thickness.

Below this deck are to be placed the propelling machinery, steering gear, magazines, shell rooms, and all that is ordinarily styled "the vitals of a war ship."

Protection of the hull againt injury to the water line region is to be afforded by means of an armor belt 3 inches in thickness, extending the length of the machinery and boiler space, and in depth from 4 feet above the 24-feet water line to 4 feet 3 inches below it.

Within this armor belt and skin plating, and above the armor, a belt, about 3½ feet wide, of cellulose is to extend the whole length of the vessel, in depth from the armor deck to the berth deck. For the length occupied by the coal bunkers, a passage 27 inches wide will be worked between the cofferdam and coal bunker bulkhead on each side; the cofferdams and passages will be suitably divided into compartments by means of transverse bulkheads.

It is intended to carry coal above the armor deck for a length corresponding to the inner bottom. This space between the armor deck and the deck above is to be subdivided by water-tight bulkheads into at least thirty-six coal bunkers, exclusive of cofferdam and passages. The space forward and abaft these bunkers is to be well subdivided by water-tight bulkheads for stores, &c.

The conning tower will have not less than 7½ inches thickness of armor, and will have a tube to the protective deck of not less than 5 inches in thickness for the protection of voice tubes, bell wires, etc.

The battery of the vessel is to be eight 8-inch B.L.R. of 35 calibers; twelve 5-inch B.L.R. rapid-fire guns; twelve 6-pounder rapid-fire guns; four 1-pounder rapid-fire guns; four machine guns; two field guns.

The 8-inch guns are to be mounted in four barbette turrets, placed one forward and one aft on center line of vessel, and one

on either side of vessel amidships. The guns in the turrets on the center line of vessel are to have a train of 310°; those in the side turrets to fire from right ahead to right astern, or to train through an arc of 180° each. The center of side turrets to be distant from the center line of vessel about 23 feet. The armor forming the barbettes, which will protect the carriages, platforms and turret machinery, is to be 8 inches in thickness for that part exposed to the fire of an enemy. Under the turrets there will be placed 3-inch armor supporting tubes, which will also protect the ammunition hoist. The armor of the turrets is to be 5½ inches in thickness, and the guns will be so mounted that they can be supplied with ammunition and loaded in any position of train. The 5-inch guns are to be protected by fixed segmental shields 4 inches in thickness. The crews of these guns are to be further protected from explosive shells by splinter bulkheads 1½ inches in thickness. Protection will be afforded the smaller guns by shields and extra side plating.

There will be five torpedo tubes, and provision will be made for defense against torpedo attacks by means of steel-ring nets carried by out-rigger booms.

The electric lighting plant will consist of three sets, each set having an engine and dynamo on the same bed plate, and each dynamo having a rated output of 400 ampères at 80 volts. The total weight of the three sets complete will be about 15 tons. The weight of the whole electrical installation, including dynamos, engines, bed plates, fittings, wirings, stores, and four search lights, will be about 40 tons.

The distribution of weights is, approximately, as follows:

Hull and fittings, exclusive of armor, . . .	3,780 tons.
Armor and protection, (including protective deck, conning tower, cellulose, &c.,)	1,850 tons.
Armament and attachments, ...	540 tons.
Machinery, stores, spare parts, &c., . . .	1,610 tons.
Coal, at normal displacement, ...	900 tons.
Equipment, outfit, stores, crew, &c., . . .	470 tons.
Normal displacement, ...	9,150 tons.

There are four triple-expansion engines of the usual vertical type, working in pairs on two shafts, disconnecting couplings being fitted, so that the forward engine on each shaft can be thrown out of service. The cylinder diameters are 32, 47 and 72 inches, and the stroke 42 inches. The estimated I.H.P. of main engines, and air and circulating-pump engines, is about 16,000 when the main engines are making 129 revolutions per minute.

The valves are all of the piston type—one for the H.P., two for the I.P., and two for the L.P. cylinders, and all are worked by Stephenson double-bar links.

The cylinders are supported upon cast-steel inverted Y columns, two to each cylinder. The engine bed plates are also of cast steel, supported on wrought-steel keelson plates, built in the vessel.

There is one cylindrical, composition condenser to each engine, with about 5,681 square feet of cooling surface, making the aggregate amount 22,724 square feet. For each condenser there will be a centrifugal circulating pump with a capacity of 8,000 gallons per minute, when drawing from the bilge, and two air pumps, each consisting of a horizontal double-acting water cylinder, driven by a single steam cylinder.

The pump casing will be continued up above the working cylinder to a height sufficient to insure the formation of a water column of such volume that the water level, in following the piston, will never fall below the top of the pump cylinder.

The pump diameter will be 16 inches and the stroke 18 inches. There will be six foot and nine delivery valves at each end of the pump, covering openings 5 inches in diameter. The valves will consist of three sheets of approved bronze, $\frac{1}{32}$ inch each in thickness, held in place by bronze spiral springs.

The steam cylinders will connect to the auxiliary steam pipe and also to a branch of the main steam pipe. They will exhaust into the main condenser, and pipes and valves will be fitted to direct the exhaust, when desired, into either of the receivers.

There are five double-ended and two single-ended boilers, the latter being intended for use when in port. The diameter of all

the boilers (external) is 16 feet 3 inches; four are 18 feet long and one 18 feet 11½ inches long, while the single-ended boilers are each 9 feet 4½ inches long. The working pressure is 160 pounds. There are four corrugated furnaces to the firing end of each boiler, or forty-eight in all, of an internal diameter of 40 inches. The total grate surface of all boilers is 1,016 square feet, and the heating surface 33,353 square feet. The shells of the boilers will be 1.39 inches thick; the stay tubes, No. 6 B. W. G. (.203″) and the ordinary tubes, No. 12 B. W. G. (.109″). The tubes are 2¼ inches external diameter.

The boilers are arranged in three water-tight compartments, each with two athwart-ship firerooms, the larger of the double-end boilers and the two single-end boilers forming the group in the middle compartment. There will be three smokepipes, 100 feet high above lower grates, and one for each group of boilers. The forced draft is on the closed fire-room system.

The disconnecting gear for throwing out the forward engine on each shaft, consists of a crank on the end of each shaft, connected by a drag link; the forward crank will have a throw of 14 inches and the after one 24½ inches. The forward crank pin will be forged or shrunk in the crank web and the after one will be fitted for being drawn back from the face of the web. When the engines are disconnected, the drag link hangs on the forward pin, and the after pin is withdrawn so as to clear the link.

The shafting is all of forged steel and hollow, and the piston and connecting rods, valve stems, links, &c., are all of forged steel. One section of line shaft for each engine will be of nickel steel, with the same external diameter as the rest, but with the axial hole enlarged so that the strength will be the same as that of the other sections.

Among the auxiliaries are two coal hoisting engines, each capable of lifting 1,000 pounds at the rate of 300 feet per minute.

The turrets will be revolved by worm gearing, connected to the bottom of the ammunition tube and driven by a steam engine. The engine will be controlled by hand wheels, situated at the sighting stations in the turrets, which, by suitable gearing, extending down the inside of the ammunition tube, operate a re-

versing valve which changes the steam and exhaust ports of the engine. The gearing will be so arranged that the turret will follow the movements of the wheel.

A change in accordance with modern practice from previous methods is the specification of metallic dial thermometers instead of mercurial ones, except for that on the main steam pipe. As the practical use of the thermometers is to indicate a correct regimen rather than any particular temperature with minute accuracy, and as the dial thermometers can be read at ten feet distance and are not liable to get out of order, it is readily seen that they will prove a great convenience, and their use represents a distinct advance. It may be added that careful tests of these instruments with standardized mercurial thermometers show that they are entirely reliable and respond quickly to changes of temperature.

There are to be two evaporators and two distillers having a combined capacity of 10,000 gallons of potable water in twenty-four hours, at a temperature not exceeding 90° Fahr.

A "dense-air" refrigerating plant, with a capacity of one ton of ice per day, will be fitted. It will include cooling pipes for the cold storage room and the scuttle butt.

The workshop will have two back-geared screw cutting lathes, the larger to swing 24 inches over ways and 4 feet between centers; a shaper of 12 inches stroke and 19 inches traverse; a power and a hand drilling machine; and a combined punch and shears. The power will be supplied by a 6 × 8-inch vertical engine.

The specifications for fitting of pipes direct that the joints between flanges in high pressure pipes shall be made with asbestos-board soaked in boiled linseed oil, "Usudurian" with copper wire gauze, "Vulcabeston," or other approved material. No material will be used that will not withstand the heat of the steam and keep tight an indefinite length of time.

The appropriation for the vessel and machinery is $3,500,000, exclusive of armament and speed premiums. In the cost of armament is included, besides the guns, ammunition, etc., all armor of turrets or barbettes, gun shields, etc., directly pertaining to the protection of the guns and loading positions.

Bids for the construction of this vessel will be opened December 15th.

Seagoing Battle-Ship No. 1.—This powerful armored vessel is somewhat larger than the three *Coast-Line Battle Ships* now building, and, as will be seen by comparing the coal capacities of the two types, is capable of keeping at sea for a considerable period.

The speed to be maintained for four consecutive hours is to be not less than an average of 16 knots an hour, and for each quarter knot maintained above that speed the contractors will receive a premium of fifty thousand dollars ($50,000.00), while for every quarter knot that the vessel fails of reaching the average speed of 16 knots there will be deducted from the contract price the sum of twenty-five thousand dollars ($25,000.00).

Length on load water line (normal displacement),	360 feet 0 inches.
Moulded breadth at load water line,	72 feet 0 inches.
Mean draught at normal displacement,	24 feet 0 inches.
Freeboard at bow,	19 feet 0 inches.
Freeboard at stern,	11 feet 8 inches.
Normal load displacement, about,	11,290 tons.
Total coal capacity,	1,975 tons.
Coal carried at normal load displacement,	625 tons.
Complement of officers and men,	486.

The hull is to be of steel, not sheathed, with double bottom and close water-tight subdivision to about 10 feet above the water line. There will be one military mast, with fighting tops, to carry no sail. The protection of the hull against injury to the water-line region is to be afforded by means of a side armor belt of a maximum thickness of not less than 14 inches and a mean depth of 7 feet 6 inches. The transverse armor at the ends of the belt will be not less than 12 inches in thickness. Barbettes and turrets for 12-inch guns will have armor not less than 15 inches thick. The ship's side, from the armor belt to the main deck, will be protected by not less than 5 inches of steel armor.

Coal is to be carried back of this 5-inch casemate armor.

An armored deck not less than 3 inches in thickness is to extend forward and aft from the ends of the armor belt.

Over the side armor belt, this steel deck will be not less than 2¾ inches in thickness. A cellulose belt is to be fitted along the sides forward and abaft the side armor belt on the 3-inch protective deck, and also on the main deck, forward and abaft the 5-inch armor.

A conning tower of not less than 10 inches in thickness, having an armored communication tube not less than 7 inches in thickness, will be carried in a suitable commanding position, the tube extending to the armor deck and affording protection to voice tubes, bell wires, &c.

The barbettes of the 8-inch gun turrets will be not less than 8 inches in thickness on the exposed side and 6 inches elsewhere. The turret armor to have a total thickness, including steel backing plate, of not less than 6 inches. These turrets will have armor-supporting tubes not less than 3 inches thick, extending from the bottom of the barbette floor to the armor deck, arranged to protect the ammunition hoists.

The 4-inch guns on the gun deck are to be protected by fixed shields 4 inches thick, built into the hull of the vessel and forming an armored sponson. The 4-inch plate is to be prolonged by armor plates 1½ inches thick, extending inboard and forming a splinter bulkhead on either side of the gun station.

Protection is to be afforded the smaller guns by shields and extra side plating.

The battery of the vessel is to be: Four 12-inch B.L.R.; eight 8-inch B.L.R.; six 4-inch rapid fire guns; twenty 6-pounder rapid fire guns; four 1-pounder rapid fire guns; four machine guns; one field gun.

The 12-inch guns are to be mounted in two barbette turrets, placed one forward and one aft on the centre line of vessel. The axes of the guns in the forward turret, when horizontal, are to be 25 feet above the load water line, and at least six feet above the deck. The axes of the guns in the after turret to be at least 17 feet 8 inches above the load water line. The guns of both turrets to be capable of firing through an arc of 270 degrees.

The 8-inch guns are to be mounted in four barbette turrets, placed two on either broadside of the vessel. The axes of the guns, when horizontal, to be about 25 feet above the load water line, and the guns in each turret to be capable of firing through an arc of about 170 degrees. The two forward 8-inch turrets to be capable of being fired at an angle of about 5 degrees across the bow; those of the two after turrets to be capable of firing at an angle of about 15 degrees across the stern.

The 4-inch guns are to be mounted as follows: Two in armored sponsons on the gun deck, capable of firing through an arc of 140 degrees, and at an angle of 5 degrees across the bow; two in armored sponsons on the gun deck amidships, between the supporting framing of the 8-inch turrets. These latter guns to fire through an arc of 140 degrees, 70 degrees forward and 70 degrees abaft the beam. The remaining two guns to be placed on the bridge at the extreme after end of the superstructure, and to be capable of firing through an arc of 140 degrees, and at an angle of 35 degrees across the stern.

The 6-pounders to be placed as follows: Two at the extreme forward end of gun deck; two on the gun deck abreast the forward 12-inch turret, and the remainder on the bridges and superstructure.

Two 1-pounders and machine guns to be placed in the military tops, and two 1-pounders at extreme after end of gun deck.

The electric lighting plant will consist of four sets, each set having engine, dynamo, and combination bed-plate, and each dynamo having a rated output of 300 ampères at 80 volts. The total weight of the four sets, complete, is about 16 tons. The total weight of the whole electric installation, including dynamos, engines, bed-plates, all fittings, wiring, and stores, and four search lights, is about 45 tons.

There will be seven torpedo tubes, and provision for resisting torpedo attack will take the form of steel ring nets carried by swinging booms.

An approximate distribution of the weights is as follows:

Hull and fittings, ...	4,540 tons.
Protection, including armor, protective deck, and water-excluding material, ...	3,630 tons.
Armament and fittings and ammunition, . .	1,000 tons.
Machinery, stores, spares, &c., ...	1,170 tons.
Coal at normal displacement, ...	625 tons.
Crew, equipment, outfit, stores, &c., . . .	325 tons.
Normal displacement, . . .	11,290 tons.

The main engines are of the same general type as in the *Brooklyn*, but there is only one set to each screw. The cylinders are 39, 55 and 85 inches diameter, and 48 inches stroke, and are expected to develop (with the air and circulating pump engines) about 11,000 I.H.P. when making 112.5 revolutions per minute. Each condenser has 8,100 square feet of cooling surface, or a total for the machinery of 16,200. There is an auxiliary condenser in each engine room, with a combined air and circulating pump.

There are five boilers, all 16 feet 9 inches diameter, two being double-ended and 19 feet long, one double-ended and 21 feet long, while the other two are single-ended and 9 feet 10½ inches long. They are placed in four water-tight compartments, the larger double-ended boiler and the two single-ended ones being in the forward compartments. The working pressure is 160 pounds.

The intermediate and low pressure cylinders are to be jacketed on the barrels and bottoms with steam from a branch connecting to the main steam pipe, reducing valves being fitted so that the jacket pressure for the I.P. will not exceed 80 pounds, and that for the L.P. 30 pounds.

The distribution valves are all of the piston type, one for the H.P., two for the I.P., and four for the L.P., all worked by Stephenson double bar links. The valve gear for all the cylinders will be interchangeable.

The boiler furnaces are 42 inches internal diameter and $\frac{9}{16}$ inch thick, of the usual corrugated pattern. There are eight for each of the large boilers and four for each of the small boilers. The total grate surface is 756 square feet, and the total heating surface 23,951 square feet.

There are two smoke pipes 100 feet high above lower grates. Forced draft is by closed fire-rooms.

The turret turning gear, workshop machinery, evaporators and distillers, refrigerating machinery and other auxiliaries are of the same style as for the *Brooklyn*.

As in the *Brooklyn*, one section of each line shaft will be of nickel steel.

The total cost of hull and machinery, exclusive of armament and speed premiums, is not to exceed $4,000,000. In the cost of armament is included, besides the guns, ammunition, etc., all armor of turrets or barbettes, gun shields, etc., directly pertaining to the protection of the guns and loading positions.

Bids for the construction of this vessel will be opened December 15th.

Olympia.—The cruiser *Olympia* was successfully launched at the yards of the Union Iron Works, at San Francisco, on November 5th. The *Olympia*, formerly known as Protected Cruiser No. 6, was authorized by the Naval Appropriation bill approved September 7, 1888, which provided for "one steel cruiser of about 5,300 tons displacement, to cost, exclusive of armament and excluding any premium that may be paid for increased speed, not more than $1,800,000."

Advertisements for bids were begun April 8, 1890, and bids were opened June 10, 1890. The contract calls for the completion of the ship by April 1, 1893.

This vessel is of a well-known type, but exceeds in size any of her predecessors in our service and the majority of such foreign vessels. The increased displacement of this vessel admits of protection hitherto unattained by this type, and allows a weight of machinery sufficient to insure a maintenance of high speed.

Length on water line, 340 feet; breadth molded, 53 feet; normal mean draught, 21½ feet; displacement corresponding, 5,500 tons; indicated horse power on trial (estimated), 13,500; guaranteed speed on trial, 20 knots, sustained sea speed, 19 knots; maximum coal capacity, 1,300 tons; complement of men, 463.

The vessel has three complete decks, including the protective

deck, and a large superstructure amidships, which is cut away forward and aft, and ends at the 8-inch gun barbettes.

She has two masts, with a fighting top and electric-light top on each, and carries sufficient fore-and-aft sail to steady her in a seaway. Her freeboard is 17 feet forward and 17 feet aft.

A complete protective deck is worked 2 inches thick on the flat throughout, $4\frac{3}{4}$ inches on the slopes amidships and 3 inches on the slope forward and aft.

A water-excluding belt 2 feet 9 inches thick is worked above the protective deck, completely surrounding the ship. It extends to a height of 4 feet above the water line. The space immediately above the protective deck is closely subdivided and much of it is covered with coal, forming an additional safeguard against the effects of damage near the water line.

The ship carries four 8-inch breech-loading rifles, ten 5-inch rapid-firing guns, twenty-four 6-pounder rapid-firing guns, six 1-pounder rapid-firing guns, four gatlings and six torpedo tubes.

The 8-inch guns are mounted on the main deck forward and aft in the center line in elevated barbettes 4 inches thick, with turrets around the guns. The latter are about 10 feet above the deck and have very great train. The armored ammunition tube is partly conical and partly straight, being 3 inches thick throughout. The 5-inch guns are mounted in the superstructure in such a way that four of them can fire right ahead, and five on either beam and four right astern. They are each protected by a 4-inch segmental shield. The 6-pounder guns have 2-inch segmental protection. There is one fixed torpedo tube in the bow, one fixed tube in the stern, and two training tubes on each broadside. The 8-inch guns are 26 and the 5-inch guns 18 feet above the water.

The engines are of the usual vertical, direct-acting, triple-expansion type, driving twin screws. The cylinders are 42, 59 and 92 inches diameter, by 42 inches stroke, and it is expected that the I.H.P., at 129 revolutions, will be 13,500, including that of the air and circulating pump engines.

All the valves are of the piston type, that of the H.P. being double ported, while the two for the I.P. and the four for the

L.P. are single ported. All are worked by Stephenson double-bar links.

The I.P. and L.P. cylinders are jacketed and the H.P. has a working liner, but will not be jacketed. The framing consists of inverted Y columns at the back, carrying the crosshead guides, and forged steel cylindrical columns at the front, supported on a bed plate of I section. As originally designed, all these parts were to be of cast steel, but owing to difficulty in obtaining the castings, the Y frames are of wrought iron, the front columns of wrought steel, and the bed plates of manganese bronze.

The crank shafts are 16 inches diameter, with $7\frac{1}{2}$-inch holes, and the crank pins 17 inches diameter, with $8\frac{1}{2}$-inch holes.

The condensers are cylindrical and of composition, having an aggregate cooling surface of 18,990 square feet.

There are six steel boilers, designed for a working pressure of 160 pounds, each 15 feet 3 inches outside diameter, with shells $1\frac{19}{64}$ inches thick; four are double-ended, 21 feet 3 inches long, while the other two are single-ended and 10 feet $11\frac{1}{2}$ inches long. The corrugated furnaces are 39 inches internal diameter, eight in the double-ended boilers and four in the single-ended ones. The total grate surface is 824 square feet, and the total heating surface 28,298 square feet. Forced draft is by closed fire rooms, and there are two smoke pipes.

The usual auxiliary pumps, evaporators, distillers, ice machines, workshop machinery, &c., are fitted.

Cincinnati.—(Cruiser No. 7.)—Was launched at the New York Navy Yard November 10th. A description of this vessel, and her duplicate, the *Raleigh*, was given in Vol. I of the JOURNAL, but a repetition will not be out of place.

They are twin screw steel cruisers, with heavy protective deck, great speed, and heavy batteries of rapid-fire guns. They have poop and forecastle decks, joined by bridges along the top of the hammock berthings, leaving an open gun deck between. Length on water line, 300 feet; beam, 42 feet; mean draught, 18 feet; displacement, 3,183 tons; sustained speed, 19 knots; maximum speed, 20 knots; I.H.P., 10,000.

The vertical keel is 36 inches deep, and .43 inch thick, while the flat keel plates are .37 and .49 inch thick. The inner bottom, worked in wake of the machinery space, is .30 inch thick. There are two longitudinals on each side of the keel below the protective deck .30 inch thick, and one above this deck .37 inch thick. The protective deck has a double slope of 22 and 39 degrees, made up of a half-inch plate all over, a 2-inch plate on the slopes amidships, reduced to 1½ inches at the ends, and a ½-inch plate on the horizontal part. There is a central longitudinal bulkhead throughout the machinery space, which, in the boiler space, forks into two bulkheads, forming a passage way two feet wide.

The bunker capacity is 556 tons, the supply on the displacement given being 400 tons.

The main battery consists of one 6-inch and ten 4-inch rapid-fire breech-loading rifles, on center pivot mounts, protected by thick steel shields, worked as part of the hull or made fast to the carriages. The 6-inch gun is mounted on the forecastle; two 4-inch guns are on the poop, two under the poop in sponsons, two under the forecastle in sponsons, and the other four, two on a side, in sponsons. The secondary battery consists of two 6-pounders, two 3-pounders, one 1-pounder R.F.G., and two 37-mm. R.C.

There are six torpedo tubes, with openings about four feet above the water, worked from the berth deck. One forward and one aft are fixed, while the other four are training tubes, and placed on the forward and after berth decks at the sides.

The rig is that of a two-masted schooner, spreading 7,210 square feet of sail. The masts have barbette galleries for machine guns just below the tops.

The engines are of the vertical, direct-acting, triple-expansion type, driving twin screws. Owing to reasons of convenience and limited room, there are two low-pressure cylinders. The cylinder diameters are 36, 53, 57 and 57 inches and the stroke 33 inches. The aggregate I.H.P. of the main engines and the air and circulating pump engines is expected to be 10,000 when the main engines are making 164 revolutions. Each engine is in a sepa-

rate water-tight compartment. The piston valves are all 20 inches in diameter, one for the H.P., two for the I.P., and two for each L.P. cylinder, all worked by Stephenson double bar links. The H.P. and I.P. cranks are opposite, as are those of the L.P. cylinders, the planes of the pairs being at right angles. The main journals are 13½ inches diameter, and the crank pins 14½ inches diameter, with 6-inch axial holes. The thrust shafts are 13 inches external and 6½ inches internal diameter, while the propeller shafts are 13¼ inches diameter, with 6½ and 6-inch axial holes. The propellers are three bladed, of strong bronze, and about 14 feet 6 inches diameter.

The condensers are cylindrical and of cast composition, 5 feet 8 inches diameter, with tubes 11 feet 6 inches long. The cooling surface of each condenser is 6,990 square feet, or for both 13,980 square feet.

The cylinders are supported on inverted Y frames, which were designed to be of steel, but there was so much trouble in obtaining steel castings that they were finally made of gun iron having a tensile strength of from 30,000 to 35,000 pounds.

There are four double-end and two single-end boilers, designed for a working pressure of 160 pounds. Two of the D.E. boilers are 13 feet 4 inches diameter, and the other two 14 feet 6½ inches; they are all 20 feet 3½ inches long. The S.E. boilers are 11 feet 6 inches diameter and 9 feet 10¼ inches long. The D.E. boilers have each six corrugated furnaces, 44 inches internal diameter for the large boilers and 40 inches for the smaller ones. The S.E. boilers have two corrugated furnaces each, 42 inches in diameter. The total grate surface is 607 square feet, and the total heating surface 20,167 square feet.

The usual auxiliaries, workshop machinery, &c., are fitted.

On September 17th, the erecting shop at the New York Yard was completely destroyed by fire and the machinery of the *Cincinnati* was damaged to an amount estimated at $17,000. This is much less than might have been anticipated, and it is expected that the completion of the ship will not be delayed.

The forged parts do not appear to have been injured at all, and the damage to cast iron portions can all be repaired by

patching and "burning." The white metal in journal bearings and eccentic straps was partially melted, and the brass condenser shells will have to be renewed.

Oregon.—The following table is of some interest in showing the number of steam engines on board a modern man-of-war. It was supplied to "Industry," a San Francisco journal, by Mr. G. W. Dickie, of the Union Iron Works, and refers to the U. S. battle ship *Oregon*:

Number of Engine Cylinders.	Purpose of Engines.	Type.	Diameter of Cylinder in inches.			Stroke in Inches.
			in.	*in.*	*in.*	
6	Main driving, 9,000 horse-power	Triple	34½,	48,	75	42
4	For air pumps	Double	6	...	...	12
4	" circulating pumps	Compound	7	12	...	6
2	Hotwell pump engines	Single	8	...	...	16
2	Fire and bilge pump engines	"	10	...	...	16
2	Air and circulating pump engines	"	10	...	...	16
4	Ventilating fans	Compound	5	9	...	6
4	Turning engines	Double	6	...	...	6
2	Reversing engines	Single	12	...	...	16
4	Hydraulic steering gear	Double	8	...	...	12
4	Main Feed pumps	Single	12	...	...	16
4	Auxiliary feed pumps	"	10	...	...	16
8	Ash hoisting	"	5	...	..	6
16	Fire-room fans	Compound	5	9	...	6
4	Steam cranes	Double	8	...	...	10
12	Hydraulic pumping	Single	20	...	...	30
8	Steam winches	Double	8	...	...	10
2	Windlass engines	"	16	...	....	12
8	Dynamo engines	Compound	7	12	...	6
2	Ice machines	Double	12	...	...	16
8	Ventilation	Compound	5	9	...	6
1	Distilling-room air pump	Single	10	...	...	12
1	Water and brine pump	"	6	...	...	10

ENGLAND.

Vulcan.—This ship represents a unique specimen of the ship constructors' art in the Royal Navy. She is known as a twin-screw torpedo depot ship, and for this purpose has been supplied with many fittings altogether unusual in her Majesty's ships. She is intended to accompany a fleet to sea, to form a depot for supplying mining and countermining stores, electrical gear,

Whithead torpedoes, &c., to assist in the work of laying out mines or fishing them up, as occasion may require, as well as forming a school of instruction for this kind of work in times of peace. She is also fitted with a factory for executing repairs to torpedo gear and boats, and to such ships of war as are not provided with the means of performing their own repairs. In addition to this, she was designed on such lines that, with the large engine power provided, she would have sufficient speed to act as a cruiser or to perform scouting work for the fleet.

She was built in Portsmouth Dockyard, her first keel plate having been laid down on June 18th, 1888, and the vessel launched on June 13th, 1889, less than a year after her commencement, a feat which was considered remarkable at that time, but has since become an affair of very ordinary occurrence. She is built of steel throughout, unarmored at the sides, but provided with a steel protective deck, varying in thickness from 5 inches in the most exposed parts to 2½ inches below the water line, the shape being the ordinary turtle-back pattern usual in cruisers of this type. This deck forms the protection for the magazines, engines and boilers, and the vital parts of the ship generally, the coal boxes also being arranged so as to materially assist in protecting these parts when they are full of coal, though naturally such a protection must necessarily be untrustworthy in a ship fitted with such powerful machinery and intended to keep the sea.

The principal dimensions are 350 feet between perpendiculars and 58 feet beam, the displacement being 6,630 tons at 23 feet mean draught. Her bunkers, twenty-one in number, stow 1,000 tons of coal, but 300 tons more can be stowed, if necessary, in the wing compartments adjoining the coal boxes, these compartments having been so fitted that coal can easily be stowed in them or got out to the fires when required. It was considered by her designers that this amount of coal would have sufficed to give the ship a cruising range of 10,000 miles at a speed of 10 knots, but sufficient allowance has not been made for the auxiliary machinery, lighting the ship, &c., and we, therefore, consider that a range of 7,000 miles is as much as she is ever likely to be

able to traverse at 10 knots speed without replenishing the bunkers.

A highly interesting feature of the ship is the hydraulic boat-hoisting machinery supplied by Sir W. G. Armstrong & Co., of Elswick. The most striking portion of this machinery, and that which has drawn forth a considerable amount of criticism, both from seamen and constructors, is a pair of enormous goose-neck cranes, placed one on each side of the ship, almost amidships. They have a total height of 65 feet, and an overhang, or rake, of 38 feet, which will allow of her boats being hoisted in and out without disarranging the torpedo net defence, a most important item in a ship of this class, the nature of the work she is designed generally to perform rendering her specially liable to sudden torpedo attack. To provide support and security for these cranes, about 30 feet of the pillar lies buried in the hold, passing down through the upper, main, and protective decks to the bottom of the ship, which is specially strengthened and fitted to support the weight and allow rotation. The upper deck, which takes the canting strain, is also specially strengthened and fitted with a heavy steel ring in which the crane rotates. Rollers are not provided to reduce the friction at this point, the crane being simply fitted with a plain disc about 5¼ inches in thickness, fitting easily in the deck socket. The lifting machinery, which is placed inside the pillar of the crane, and therefore, being well down in the hold, is fairly protected from gun fire, consists mainly of two hydraulic rams—the larger one 17½ inches in diameter, the smaller one 5½ inches in diameter—having a vertical stroke of 10 feet, giving a lift of 40 feet at the purchase, through the multiplying power of the fourfold pulleys, which the rams are constructed to operate in the usual manner. At the ordinary working pressure of 1,000 pounds on the square inch, the thurst on the crosshead carrying the pulleys is about 118 tons, which gives a lifting power at the jib of 20 tons, moving at the rate of 90 feet per minute. The small cylinder plays an important part in this mechanism, being designed to mitigate the danger which always attends the hoisting in or lowering of boats in a seaway, and is the subject of a patent by the firm. Its

special duty is to keep taut the slings after they have been hooked on to the boat, until a favorable moment for hoisting presents itself. The valves are so arranged that the main lifting power cannot be applied until the small cylinder has been brought into action, the relation of power to weight being so adjusted at the proper working pressure that the slings are only held taught while the boat is rising or falling in the waves, provision being made for the escape of the water from beneath the ram as the weight of the boat forces it back against the pressure. The cranes are revolved by means of a pair of rams 16 inches in diameter, placed vertically alongside the pillar of the crane. It has been necessary to provide considerable more power for this purpose than would be necessary for a similar crane on shore, since it will be obvious that if boats are to be hoisted in a seaway, it is necessary to have sufficient holding power to overcome the additional twisting strains brought to bear when the ship is rolling. The direct thrusting power of the rams is 90 tons, and they operate a 2⅛-inch chain cable, which passes round and is secured to a steel sprocket drum at the foot of the crane 6 feet 6 inches in diameter, thence over a pulley at the head of each ram, the ends being secured to the framing which supports the ram cylinders. The cranes have a turning range of about 250 degrees, and the boats are so placed as to come fairly within the scope of one or other of them; but, since the *Vulcan* is 58 feet wide, and the cranes 40 feet apart, it is obvious that all the boats cannot come within the scope of either crane. To provide for the contingency of one of the cranes be coming disabled, the torpedo boats—of which the ship carries six—are stowed in crutches fitted on trolleys, which may be traversed from one side of the ship to the other. It is anticipated that when the ship is in commission, and the crew properly drilled, the whole of these boats may be lifted in or out in less than a quarter of an hour.

An ordinary derrick attached to the mainmast for boat-hoisting purposes is also fitted with similar hydraulic gear for topping the derrick and lifting the boats, the lifting power in this case being 10 tons only. In addition to the cranes and derricks,

there are also four hydraulic bollards, or winches, provided for general hauling purposes, and for use with the derricks. They are guaranteed to be equal to a direct pull of 30 cwt.

The boat equipment consists of six second-class torpedo boats, a wood vedette boat 53 feet in length, a 42-foot steam pinnace, and a steam cutter. The torpedo boats are built entirely of steel, 60 feet in length, having a speed of over 16 knots per hour, the engines working up to about 430-horse power, the boilers being of the ordinary locomotive type, working under forced draft with closed stokeholds. The vedette boat and steam pinnace are also fitted to work with closed stokeholds and forced draft. In addition to these boats, the ship also carries two specially designed countermining launches, besides the ordinary equipment of sailing and pulling boats.

For offensive purposes the ship carries an armament consisting of eight 4.7-inch rapid-firing guns, mounted upon the Vavasseur central pivot system, four forward and four aft on the upper deck. These are fitted with the usual protective shields, and have a good arc of training, the bow and stern guns both firing across the keel. In addition to these there are twelve 3-pounder quick-firing guns distributed between the main and upper decks. In addition to these guns the ship will be provided with an equipment of thirty Whitehead torpedoes, some of which will be of the latest pattern, 18 inches in diameter, carrying a charge of 242½ pounds of dry gun-cotton, and having a total weight—including the head—of upwards of 1,150 pounds, the remainder consisting of the ordinary pattern torpedo, 14 inches in diameter, and carrying various charges up to 115 pounds of gun-cotton. For discharging these torpedoes, two fixed tubes are provided below the water line forward, one for 18-inch and one for 14-inch torpedoes; a fixed tube at both bow and stern for 14-inch torpedoes, above the water line, and two broadside training tubes for 14-inch torpedoes above the water line aft.

In addition to the locomotive torpedoes, the ship also carries a large number of mines, which may be laid down as desired. Some of these carry as much as 500 pounds of gun-cotton, the total weight of each mine when complete reaching to upwards

of 1,300 pounds. Last, but not least, of the offensive weapons of the ship, is the powerful ram with which she is provided.

The ship is lighted throughout by electricity, and fitted with four electric search lights, each of 25,000 candle power. For these purposes three sets of Siemens dynamos, driven direct by Willans' patent compound central valve engines, are provided, each dynamo giving a current of 400 ampères at 80 volts E.M.F. when driven at about 420 revolutions per minute.

Being unarmored, great speed is relied upon as the principal defence, but she is also provided with a torpedo net defence, besides the protective deck metioned above, while in addition the principal of subdivision has been thoroughly carried out in her construction, in order to minimize as much as possible the chances of her being sunk in action by a lucky hit from either torpedo or shot, the number of these water-tight compartments reaching to upwards of 140. The carrying out of this principle of minute subdivision into water-tight and air-tight compartments brings into greater prominence the matter of ventilation, a proper supply of fresh air to these compartments being of course necessary to enable the work of the ship to be carried on without detriment to the health and comfort of the crew. To insure this, numerous air shafts are led down from the upper deck, some for supply and others for exhaust, air trunks being led from these to every compartment in the ship, including bunkers, magazines, store-rooms, &c., ventilating fan engines driven by steam being connected to some of the most important trunks to supplement the natural draft when necessary. To preserve the principle of water-tightness it becomes necessary to fit each of these air openings with a valve capable of being shut down when necessary, and when it is remembered that many compartments have a valve for supply and another for exhaust, some idea may be formed of the enormous number of valves necessary for this purpose alone; when to these are added the water-tight doors and hatches, the sluice and drain valves, and suction valves to the various pumps, mostly fitted with shafting and gearing to admit of their being worked from the main deck, it will be seen

that the task of keeping all in good working condition forms no small portion of the duties of the engineering staff of the ship.

The propelling machinery and boilers were supplied by Messrs. Humphrys and Tennant, who, indeed, were responsible for the supply and fitting of all the auxiliary machinery in the ship, with the exception of the hydraulic machinery. They also supplied the fittings for the factory except the machines, but all machines, shafting, &c., were fitted in place by their workmen. The main engines, of which there are two sets driving twin screws, are of their well-known type of triple-expansion overhead cylinders, capable of developing 12,000 horse power, at 100 revolutions per minute. The cylinders are of 40-inch, 59-inch and 88-inch diameter, and the stroke 4 feet 3 inches. The air pumps are driven direct off the low-pressure pistons, and the condensers are of gun-metal, giving a cooling surface of 13,500 square feet. The circulation of the cooling water through the condenser is maintained by independent centrifugal pumps, driven by Humphrys' single-cylinder overhead engines. These pumps are duplicated, to provide for the contingency of one becoming disabled, the old plan of fitting sea injection cocks in case of emergency having been discontinued. These pumps are also fitted to draw from the bilge, and guaranteed to take 1,000 tons per hour from the bilge with 100 pounds pressure of steam. This they easily did at the trial, the total amount of water delivered by the four engines, as ascertained by experiment, being a little over 5,000 tons per hour. There are no feed pumps driven off the main engines, but separate feed engines, two main and two auxiliary, all of equal size and Admiralty pattern, are fitted in the stokeholds for supplying the main boilers, a smaller pattern being supplied for the auxiliary boiler. In each engine-room there are also two fire and bilge engines, one turning and one reversing engine, a distilling pump, an auxiliary condenser, which takes the exhaust steam from the whole of the auxiliary service, having its own circulating engine and air pump, and a Normandy's double-distilling apparatus capable of providing 150 gallons of fresh water per hour for the use of the crew.

Separate engine rooms are provided for the hydraulic engines,

of which there are two complete sets, each set equal to the task of supplying the pressure for the cranes and bollards, &c., when hoisting in the boats. They are of the horizontal tandem compound type, two high-pressure cylinders of 13½ inches diameter and two low-pressure cylinders 27 inches diameter, with a stroke of 18 inches, to each set, the pumps, 5¾ inches diameter, being driven direct off the pistons in the ordinary way. No hydraulic accumulator is provided, but a hydraulic governor efficiently supplies its place by controlling the supply of steam to the engines as the water pressure varies. Murdock's steam governor is also connected to the throttle valve in case of mishap to the pump, to prevent the engines running off at a breakdown speed. In each hydraulic engine room there is also one of Belliss' air compressing engines and pumps, with reservoir for supplying Whitehead torpedoes with air up to a pressure of 1,700 pounds per square inch, besides ventilating and exhaust fans driven by Brotherhood's three-cylinder engines; and, on the starboard side only, one of the dynamos—termed the war dynamo, because it is placed under the protective deck—the other two being placed in the workshop above the protective deck, where it is presumed they can be much better attended to, being under the direct supervision of the officer in attendance there.

In the workshop there are five lathes of various sizes, from 15-foot bed and 9-inch centers down to 3-foot 6-inch beds and 6-inch centers, two drilling machines, planing, slotting, shaping, and punching machines, a circular saw-bench, a carpenter's bench and fitters' benches, and a Fletcher's air furnace capable of melting down 2 cwt. of scrap steel in two hours. The pulley shafting is driven direct by a 6-inch belt off a small single-cylinder horizontal engine of Messrs. Humphrys' design and construction. In the blacksmiths' shop on the upper deck is a powerful hydraulic forging press, a large forge, fitted with Root's blower and steam blast, also a coppersmith's forge and pipe-bending machine, together with a complete set of tools.

To supply steam there are four double-ended cylindrical boilers, 17 feet long and 14 feet in diameter, with three furnaces each end, 3 feet 8½ inches diameter, 6 feet in length, leading to a com-

mon combustion chamber. There is also, in addition to these, a smaller boiler, single-ended, 12 feet 5 inches in diameter and 9 feet 4 inches in length, with three furnaces, which is intended to be used generally for auxiliary purposes, but which is worked under the same conditions of forced draft as the main boilers. The stokeholds are closed, and fitted in the usual way with air locks for entrance, air-tight hatches, &c. The air for the fires is driven into the stokeholds by eight 60-inch fans, driven by Brotherhood's three-cylinder engines working up to 600 revolutions per minute. As originally designed, the total grate area was about 655 square feet, and the total heating surface about 19,000 square feet. Certain modifications, however, were suggested by experience with similar boilers in other ships, in order to prevent overheating of the tube plates, and consequently leaky tubes. These when carried out reduced the grate area to about 543 square feet, and the heating surface to 16,740 square feet. In spite of these alterations, however, some trouble has been experienced with leaky tubes when the boilers are cooling down after forced draft has been applied; and it was at one time thought that new boilers would be necessary. More recent experience with some ingenious devices of Messrs. Humphrys and the Admiralty engineers has, however, led to the expectation that all these troubles may eventually be overcome, and the commissioning of the ship has therefore been delayed until the alterations suggested by the success which attended the recent trials of H. M. S. *Thunderer* at sea have been carried out. It is understood that the boiler tubes are to be fitted at the combustion chamber ends with the Admiralty ferrule, and the heating surface and grate surface restored to the amounts approximately as designed. It has also been decided to fit additional evaporators in order to maintain the supply of fresh water for the use of her own and the torpedo boats' boilers. Grease extractors are also to be fitted to prevent the admission of oil into the boilers. We cordially wish success to the Engineer-in-Chief of the Navy in his laudable attempt to save the taxpayers' pockets. The weight of the machinery is 1,030 tons, or 11.6 indicated horse-power per ton.

Curiously enough, in the first trial trips, which were carried

out to determine the acceptance of the machinery by the Admiralty from the contractors, no trouble was experienced with the tubes, a mean speed of 18.5 knots having been maintained for eight hours with a horse-power of 8,167, the revolutions of the engines being 88.6 starboard and 88 port, with a mean air pressure in the stokeholds of .457 inch of water. It has not yet been considered advisable to attempt the 12,000 horse-power; but it is to be hoped that this magnificent vessel may not have to be kept waiting ignominiously in harbor much longer on account of boiler-tube troubles, but that with more modern devices she may be enabled to attain to her full speed of 20 knots for a few hours, and to keep her sea-going speed of 18 knots as long as her coal may last.—"The Engineer" (London).

Barfleur.—On the afternoon of August 10th the first-class ironclad *Barfleur* was successfully launched from Chatham Dockyard.

She is the largest and heaviest ship which has yet been launched there. She was laid down in October, 1890, and was originally designed as a second-class line-of-battle ship; but as her armament will in disposition, if not in weight, be the same as that of a first-class ship, she is now ranked in the latter class. The construction of the vessel has been completed with remarkable rapidity, and the cost of construction has been much reduced. The *Barfleur* is the sister ship of the *Centurion*, recently launched at Portsmouth. The total length of the ship is 378 feet; breadth, 70 feet; draught of water, 25 feet 6 inches; weight, when ready for sea, 10,590 tons; and weight, when launched, 5,200 tons. She is built entirely of steel. The sides of the ship, in the vicinity of the load water line, are covered with steel-faced armor of thicknesses varying from 9 inches to 12 inches. Between the top of this armor and the main deck the sides are covered with nickel-steel armor 3 inches thick. The bottom of the ship is covered with teak 4 inches thick, to a height of 4 feet above the load water line. She has two barbettes, each of which carries two 10-inch guns. These are protected by armor 9 inches, 8 inches and 5 inches thick on the vertical sides of the barbettes. She has two sets of engines, capable of developing

13,000 H.P. and of giving her a maximum speed of 18¼ knots. The total amount of coal which can be carried is 1,240 tons, and this will enable her to steam for three days continuously at full speed, or for 21½ days continuously at ten knots, the distance traveled in the latter case being 5,180 knots. The engines, boilers, and also the magazines, are protected by a steel deck 2 inches in thickness. Each of the four 10-inch guns is capable of throwing a shell weighing 500 pounds, or 4½ cwts., a distance of six miles, the charge of gunpowder required to fire this projectile being 252 pounds. In addition to these large guns, the ship is provided with ten 4.7-inch quick-firing guns, each of which is capable of throwing a shell weighing 45 pounds a distance of five miles, the gunpowder charge required being 12 pounds. Each of these latter guns can fire 14 shells per minute. Several other smaller guns are to be fitted at various parts of the ship, including 3 and 6-pounders and Nordenfelts. The ship is also fitted for projecting torpedoes. The crew will consist of an Admiral and 630 officers and men.

In the launch of the *Barfleur* a new and, as the event proved, a very successful departure was made from the plan usually adopted for securing the poppetts that support the fore end of the ship. The new device provides for the fitting of plates from side to side, under-running the keel. Four wide strap-plates, 1 inch thick, were fixed in position, the foremost pair being riveted together, with two large angle-bars at the fore-edge to give additional security. The upper ends of the strap-plates were tied together by longitudinal plates, to which were attached the angle-irons for receiving the heads of the poppetts. Thus the fore part of the ship was suspended as in a sling; and by this arrangement—working a strake or two of planking slightly thicker than usual amidships—the necessity was obviated of perforating the planking of the hull in any way. This plan was tried for the first time in connexion with the launch of any ironclad, and the experiment was completely successful.

The *Sappho*, built by Messrs. Samuda Bros., and engined by Messrs. John Penn & Sons, underwent her trials off Sheerness in the latter part of September. A preliminary run was made on

Tuesday, the 20th of September, for the purpose of testing the anchor and steering gear, &c., when, everything being found satisfactory, the vessel was put on her eight hours' natural draft full power trial on the following day. The mean results were as follows: Steam, 137; vacuum, starboard, 26.78, port, 28.21; revolutions, starboard, 128.25, port, 129.81; mean pressure, starboard, high, 44.66; intermediate, 23.75; low, 13.41; port, high, 51.67; intermediate, 25.4; low, 12.3; I.H.P., starboard, 3,555.47; port, 3,745.8—making collectively, 7,301. The maximum collective I.H.P. obtained was 7,689. The boilers steamed with great freedom, with an air pressure in the stokeholds of $\frac{3}{10}$ of an inch. On Thursday the boilers were examined and tubes swept, and the four hours' forced-draft full-power trial took place next day, with results considered in all respects most satisfactory, both as regards speed of ship and power developed by the machinery. The engines worked with perfect smoothness during the whole of the trials, and without the least hitch of any kind. The results attained were as under: Steam, 142.6; vacuum, starboard, 27.7, port, 27.7; revolutions, starboard, 140.16; port, 140.5; mean pressure, starboard, high, 56.4; intermediate, 28.55; low, 15.29; port, high, 62.2; intermediate, 30.6; low, 15.3; I.H.P., starboard, 4,671.65; port, 4,945.84—making collectively 9,617.5 I.H.P. The maximum I.H.P. realized was 9,861. The boilers maintained steam at $\frac{7}{10}$ of an inch of air pressure in the stokeholds, there being no sign of leakage in any part during the trials, nor after examination of boilers when opened out after the trials. The speed of the ship, as ascertained by log, was 19.41 knots for natural draft and 20.47 knots at forced draft, the ship drawing 14 feet forward and 17 feet 6 inches aft. Advantage was taken of the early completion of the machinery trials to make the usual tests of the gun mountings, &c., which were found in all respects satisfactory.

Blenheim.—Full details of the forced-draft trials of the new first-class cruiser *Blenheim*, built by the Thames Ironworks Co., and engined by Messrs. Humphreys, Tennant & Co., were received at Sheerness on September 21st. The H.P. required under the contract was 20,000, which was exceeded, the mean for

the four hours' run being 21,411 H.P., this being obtained with the engines averaging 105 revolutions per minute. The speed by the patent log for the four hours' run was 86.4 knots, giving an average of 21.6 knots. The engines and boilers worked very successfully throughout the trial.

Brilliant.—The new second-class cruiser *Brilliant*, built at Sheerness Dockyard under the Naval Defense Act, underwent her trials off Sheerness during the first week in October. The Admiralty was represented by Mr. R. J. Butler, and the machinery contractors, Messrs. R. and W. Hawthorn, Leslie, and Co., Limited, of Newcastle-on-Tyne, by Mr. F. C. Marshall, under whose supervision the trials were conducted. In each case the draught forward was 15 feet and aft 18 feet 6 inches. The mean results of the natural draft and forced draft trials were as follows:

	Natural Draft.		Forced Draft.	
Steam	147.8		146.7	
	Starboard	Port	Starboard	Port
Vacuum	26	26.09	25.5	25.5
Revolutions	134.1	132.5	142.8	141.4
Mean pressure. High	45.7	44.5	49.7	51.8
Mean pressure. Intermediate	28.4	26.5	32.1	30.8
Mean pressure. Low	12.1	11.8	13.5	14.1
Indicated horse-power. High	1064.4	1023.9	1232.6	1273.0
Indicated horse-power. Intermediate	1414.9	1307.7	1702.8	1622.6
Indicated horse-power. Low	1376.4	1325.3	1634.2	1698.4
Total indicated horse-power of starboard and port	7512.6		9163.6	
Air pressure	.16		1.1	
Speed of vessel	19.2		20.4	

Owing to the coal being of an inferior quality a higher air pressure was found to be necessary than on the trials of the sister ship H. M. S. *Sybille*. As shown above, the speed of the *Brilliant* under both natural and forced draft is in excess of that anticipated by her designers, the official estimate being 18.25 knots under natural draft and 19.75 knots under forced draft. During both trials the engines worked with perfect smoothness, and the boilers generated a plentiful supply of steam.

Torpedo Gunboats Leda, Alarm, Onyx.—Two torpedo gun-

boats, named respectively *Leda* and *Alarm*, were recently floated out of No. 1 dock at Sheerness, while a third was christened *Onyx*, on leaving the slipway at Messrs. Laird's establishment, Birkenhead. These three vessels belong to a class officially recognized as improved *Sharpshooters*. They are alike in every detail, and measure 230 feet by 27 feet, with a normal displacement of 810 tons. Their engines are to develop 3,500 H.P. under forced draft, propelling them at a speed of between nineteen and twenty knots. Considering their size, the armament to be carried by them will be of the most powerful description, and will include two 4.7 inch and four 3-pounder quick-firing guns, in addition to five torpedo tubes. The machinery of the *Leda* and *Alarm* will be supplied by Messrs. Penn & Sons, while the *Onyx* will be engined by her builders. The estimated cost of these vessels is as follows: *Leda* and *Alarm*, $290,000 each; and *Onyx*, $275,000.

Dryad.—A new torpedo gunboat of a similar type to the *Leda*, but of larger dimensions, has just been commenced at Chatham. She is to be named the *Dryad*, and will measure 250 feet by 30 feet. Her displacement will exceed that of the *Leda* class by 260 tons, and she will be provided with a poop and forecastle. The contract for supplying the engines of this vessel has been given to Messrs. Maudslay, Sons & Field, London. They are to be of 3,500 H.P. and are estimated to propel her at a speed of nineteen knots. The *Halcyon*, *Harrier*, *Hussar* and *Hazard*, sisters to the *Dryad*, are to be built in the Royal dockyards, the first three at Devonport and the latter at Pembroke. The machinery for the trio to be constructed at Devonport will be supplied by Messrs. Hawthorn, Leslie & Co., Newcastle-on-Tyne; each vessel will be supplied with two sets of vertical triple-expansion engines, having cylinders of 22-inch, 34-inch and 51-inch diameter, with a stroke of 21 inches. Steam will be supplied from four locomotive boilers at a pressure of 155 pounds, and under full power the engines will run at 250 revolutions per minute.

Jaseur.—On Saturday, September 24th, the Naval Construction and Armaments Co. launched from their yard, at Barrow,

the *Jaseur*, the second of three fast twin-screw steel torpedo gunboats, which they are building for the British Navy. The *Jaseur* is one of six gunboats, three of which are being built at Barrow, and which complete the scheme under the Naval Defense Act of 1889. She is 230 feet between perpendiculars, 27 feet extreme breadth, 14 feet 3 inches depth, moulded, with displacement of 810 tons at 8 feet 9½ inch load draught. She is propelled by twin screws, with two sets of independent triple-expansion vertical inverted cylinder engines capable of developing 3,500 I.H.P., and driving the vessel at a speed of 20 knots, the diameters of the cylinders being 22 inches, 34 inches and 51 inches, with 21-inch stroke. The engines are placed in separate engine-rooms enclosed by watertight bulkheads. This class of gunboats has been designed for high speed. The engines and boilers are protected from shot and shell below the water line by the coal bunkers with 3 feet thickness of coal. She will be fitted with a powerful search light and a rifle-proof conning tower. The armament consists of six guns and three torpedo tubes to discharge Whitehead torpedoes.

New Torpedo Boats.—The immediate construction of fourteen torpedo boats has been ordered. These boats are of two classes —a class of ten torpedo-boats, larger and faster than any at present in the British Navy; and a class of four vessels which, while larger and more seaworthy than any of the present torpedo boats, are considerably smaller, yet much faster, than any of the torpedo gun vessels. The building of the ten torpedo boats was foreshadowed in the last Naval Estimates; the building of the larger vessels has been determined more recently. Of the torpedo boats, three have been entrusted to Messrs. Yarrow & Co., of Poplar; three to Messrs. Thornycroft & Co., of Chiswick; three to Mr. J. Samuel White, of Cowes; and one to Messrs. Laird Brothers, of Birkenhead. Two of the larger vessels are being undertaken by Messrs. Yarrow and two by Messrs. Thornycroft.

The ten torpedo boats measure 140 feet in length by 14 feet 6 inches in breadth; and they are therefore 10 feet longer and 1 foot broader than the six boats of the class which was added to

the Navy in 1889, and which at present constitutes the only modern class in the British Navy. The boats of 1889 had a guaranteed speed of 22.5 knots, and, with a load of 20 tons on board, did a little more than that during a continuous run of three hours. The new boats are to have a guaranteed speed, with a load of about 25 tons, of 23 knots; and some of the builders are confident that they will have an actual speed, in smooth water of 23.5 knots, or just over 27 statute miles an hour.

The four remaining boats, which may be provisionally described as "division boats," are vessels which, when completed, will have their equals in few navies and their superiors in none. They are to measure 180 feet in length and 18 feet 6 inches in breadth, and the speed expected of them is not less than 27 knots, or just over 31 statute miles an hour. Though very little larger than some torpedo boats, and offering, therefore, a scarcely more conspicuous target to projectiles or torpedoes, they will possess an advantage that none of the existing "catchers" enjoy, in that they should be able to outstrip all catchers and to overhaul all, or very nearly all, torpedo-boats.

For the sake of comparison, brief details of some of the most modern foreign craft of nearly corresponding classes are appended:

	Length.	Beam.	Extreme Speed.
Torpedo Boats:			
Schichau's *Adler* class	152.5 ft.	16.7 ft.	27.4 knots.
Normand's *Dragon* class	137.7 ft.	14.7 ft.	25.0 "
Thornycroft's *Ariete* class	135.0 ft.	14 0 ft.	25.0 "
Yarrow's *Pinero* class	130.0 ft.	13 6 ft.	24.4 "
New British class	140.0 ft.	14 5 ft.	23 5 "
Division Boats:			
French *Leger* class	197.0 ft.	23 0 ft.	18.5 "
German *D.* 7 class	192.0 ft. (?)	23 5 ft. (?)	26.0 "
New British class	180.0 ft.	18 5 ft.	27.0 "

Mere figures do not, it is believed, convey a fair idea of the capabilities of the new British torpedo boats, for it is expected that, owing to their design, they will, in a sea way, be nearly as

fast as any foreign boats, though in smooth water they will be considerably slower than some. Yet it might be wished that greater speed were promised with them. The particulars of the new "division boats" speak for themselves, and seem to leave little or nothing to be desired.

Light Draught Paddle Steamer.—The British Admiralty have entered into a contract with Messrs. Yarrow & Co., of Poplar, for the immediate construction of a small paddle steamer, to have the least possible draught practicable, for the navigation of the Upper Shire. This vessel is destined to maintain communication at all times of the year between Lake Nyassa and Matope, where the navigation terminates, on account of a succession of rapids. From there the communication is carried on by land porterage over the highlands, as far as Katunga, on the Lower Shire, this being the highest point which the gunboats *Mosquito* and *Herald* can reach. This little steamer is 60 feet in length by 14 feet beam, and will have a draught of 11 inches, and speed of from eight to nine miles an hour. She will be put together on the Upper Shire, having been first transported there in loads, mostly of a size suitable to be carried by one or two negroes, and when in regular work will act in conjunction with the two gunboats, *Pioneer* and *Adventure*, which Messrs. Yarrow & Co. are constructing for the Government for service on Lake Nyassa.

Stern-wheel Gunboat for the Upper Chindwin.—The Indian Government have placed an order with Messrs. Yarrow & Co., of Poplar, for the construction of one of their shallow-draft steamers, for service on the upper part of the Chindwin, in Burma. The various military police posts on this river are at present without any reliable means of maintaining communication, and it is to meet this want that the steamer in question has been ordered. The Chindwin is a river most difficult to navigate. It abounds in shallows, rocks and sandbanks, and for this reason the stern-wheel type was selected as the class of vessel most suitable to conform to the conditions necessary. The vessel is to have a length of 90 feet, beam of 18 feet, draught of 13 inches, and a speed of between eleven and twelve miles an hour. The steamer will be arranged so that if necessary a number of

machine guns can be mounted on the upper deck, with a view to secure a good range over the river banks in the event of a disturbance with the natives.

Success of Ferruled Boiler Tubes in the British Navy.—The return of the *Thunderer* from her protracted steam trial to Madeira has removed all doubt with regard to the efficacy of the new patent ferrule, as applied to boiler tubes. The passage out and home was performed at an average speed of twelve knots, and throughout the trial the boiler tubes showed no sign of leakage. In fact, the highly successful results of the trial will, no doubt, lead to the universal practice of ferruling the boiler tubes in all our war vessels fitted with forced draft. As it is, the boilers of the torpedo depot-ship *Vulcan* and the second-class battleship *Devastation* are to be treated similarly to those of the *Thunderer* forthwith, while the Clyde-built cruiser *Iphegenia* is at present undergoing the process of ferruling at Portsmouth. The decision to carry this out in the latter vessel was arrived at after her recent failure to pass through the eight hours' natural draft trial of her machinery. At the end of the sixth hour considerable trouble was experienced with No. 3 boiler through excessive leakage of tubes, and the trial had to be abandoned. The average results obtained, however, during the six hours that she was under steam were remarkably satisfactory, the speed maintained being 19.6 knots with 7,445 H.P., or 445 H.P. above that contracted for. After the boilers are fitted with the new ferrule, the vessel will be again put on trial preparatory to the acceptance of her machinery by the admiralty.

The whole of the new boilers, twelve in number, of the second-class battle ship *Temaraire*, which is now undergoing a thorough overhaul at Devonport, have been placed on board. These boilers were constructed at the dock yard there, and weigh, on an average, 30 tons each, their normal working pressure being 120 pounds per square inch.—"Marine Engineer."

Mishaps during the 1892 Manœuvres of the British Navy.—As far as breakdowns and mishaps are concerned, there seems to have been a little reality after all in the naval manœuvres just brought to a close. The frailty of the torpedo boats and torpedo gunboats

has been much in evidence during the mimic war, and the experiences of their commanders have been rather unenviable. The unlucky *Sharpshooter* was again in trouble, and her boilers seem to have developed more or less serious defects which will require overhauling at Devenport. While in the Bristol Channel she experienced very bad weather, and shipped some heavy seas; in fact, it is stated that at one time she had upwards of seventy tons of surplus water on board, and that her engineer and his men were working in the stokehold up to their waists in water. The fires were eventually put out, after which she remained for a considerable time in danger, but was finally towed to Plymouth by the cruiser *Thetis*. The *Spanker*, a sister vessel to the *Sharpshooter*, which failed to undergo her commissioning trial for the manœuvres with satisfaction, through defective boiler stays, gave her engineers some trouble and anxiety during the war. The crown of one of her boilers became so badly bulged that she had to retreat to Milford Haven for repairs. It was found, however, that the defects were too serious to be dealt with, and the vessel was ordered to Devonport for a thorough overhaul of her boilers. A more serious accident than that alluded to in the *Spanker* occurred on board torpedo boat No. 25. While off Queenstown the crown of the boiler gave way, and the flames belched forth into the stokehold, seriously burning three men. From a monetary standpoint the most serious accident that occurred during the manœuvres was the sinking of torpedo boat No. 75, which resulted after a collision with No. 77 off the Maiden's Rock. The total value of this vessel may be set down at $8,000, after making some deduction for depreciation on the first cost. She was fitted with engines capable of developing upwards of 750 H.P., and maintaining a speed of 18 knots. At the time of her loss she carried one machine gun, two Whitehead torpedoes, and an electric search-light. She measured 125 feet in length, and had a displacement of about 66 tons.

FRANCE.

The New Armored Cruisers.—Stanislas Dupuy de Lôme was the designer of *La Glorie*, a vessel that may fairly be considered

the earliest of the sea-going ironclads, and the name, therefore, of the great naval constructor has been given, not inappropriately, to the first of an absolutely new type of armored cruisers destined for the service of France.

The *Dupuy de Lôme*, built at Brest from the plans of M. de Bussy, was launched so long ago as October 27th, 1890, but is only now approaching completion for regular service. She differs from all previous ships of her class in that she is fitted with three screws, and from all modern ships of her class in that she is covered with what is practically a complete coating of armor. There are now building five other cruisers which partake of some of her leading peculiarities, and as she is of a type which cannot but prove exceeding formidable, a description of her can scarcely fail to be of interest. Constructed entirely of steel, she is 374 feet long, and 51.5 feet broad, and has a depth of 35.7 feet. With a draught of water aft of 24.6 feet, and a mean draught of 23.4 feet, she displaces 6,300 tons of water, and she is, therefore, a good deal smaller than our new unarmored cruisers of the *Edgar* class. Her great length, in proportion to her beam, is especially noticeable. The length of ships of our *Orlando* class is only 5.33 times their breadth; of ships of our *Blake* class, only 5.77 times their breadth; and of ships of our *Edgar* class, only 6.0 times their breadth; while the length of the *Dupuy de Lôme* is 7.26 times her breadth. Her proportions, indeed, are almost like those of a torpedo-boat catcher, and they are calculated greatly to assist her speed. With engines indicating in the aggregate 14,000 H.P., when worked under forced draft, she will make 20 knots, and, working with natural draft, she will make 17.5 knots. Like nearly all French war ships of large size, she tumbles home considerably all round, and particularly so at the bows, where her form is that of the toe of a boot; but she has a high freeboard, and has every appearance of being a comfortable craft. She is plated from well below the water line to the level of the upper deck with steel armor 3.94 inches thick. In addition, she has, near the water-line level, a curved end-to-end protective deck of 2 inches steel; and below this, over the boilers and engines, there is a splinter deck, the space between the two being filled

with coal. There are, as in nearly all modern cruisers, side cofferdams which will likewise be filled with coal; and there are also thirteen principal and numerous smaller water-tight compartments. It has been found by experiment that the 3.94-inch side armor is sufficient to cause 5.5-inch and 6.2-inch shells, charged with mélinite, to burst outside; and, seeing that comparatively few fast cruisers carry many guns of larger calibers than these, and that, at present, the largest quick-firing gun that has been anywhere adopted is of about 6 inches caliber, it is anticipated that the *Dupuy de Lôme*, in action with a cruiser, will not be much inconvenienced by the bursting between decks of high explosive projectiles. This is a most important consideration, for the *Resistance* experiments seem to have established as a fact that shells bursting between decks will do nearly as much damage to human life by their gases as by their fragments.

The arrangement of the *Dupuy de Lôme's* armament is peculiar. Just forward of amidships on each broadside there is on the upper deck a sponson supporting a 7.46-inch, 11-ton breech-loading gun (model 1887) of 45 calibers. Each is covered by a revolving 3.94-inch armored turret that is moved by hydraulic machinery. Between these turrets, and along the greater part of the deck, runs a light superstructure, which is very sharp forward, and at its extreme end, slightly raised above the superstructure deck, is a 6.2-inch, 6.5-ton quick-firing gun of 45 calibers. Two more guns of the same model are placed on a somewhat lower level, in such a manner that one commands each bow. All these guns are in separate turrets, as before. At the stern the arrangement is similar, save that the three 6.2-inch quick-firing guns are all on the superstructure deck. In consequence, the ship can fire simultaneously, either right ahead or right astern, two 7.46-inch and three 6.2-inch guns; and on either beam, one 7.46-inch and four 6.2-inch guns. There are two large military masts of steel, and in the interior of each is a winding staircase, which leads to the lower top. These masts, which are pierced, will serve in action as conning towers. Each carries in its lower top two 3-pounder quick-firing guns, and in its upper top an electric search-light projector. A little abaft of the forward group of 6.2-inch

guns there is on each bow a 6-pounder quick-firing gun. A little forward of the after group of 6.2-inch guns there is on each quarter a 6-pounder quick-firing gun; on the bridge there are two Hotchkiss 1-pounder revolving cannon; and on the superstructure forward there are two more guns of the same kind, so that, when chasing, the cruiser can bring to bear two 7.46-inch, three 6.2-inch, two 6-pounder, two 3-pounder, and four 1-pounder guns. All the pieces have an initial velocity of 2,400 foot-seconds or upwards, and possess, therefore, great power in proportion to their caliber. The 6.2-inch quick-firing guns are converted breech-loading guns, model 1887.

Similar in general design to *Dupuy de Lôme*, but somewhat smaller than she, are the four ships, *Charner* and *Bruix* (building at Rochefort), *Chanzy* (building at Bordeaux), and *Latouche Treville* (building at Le Havre). Larger than these, yet still smaller than the *Dupuy de Lôme*, is the sixth cruiser of the series, the *Pothau*, which is to be begun this year in a contractor's yard. The *Charner* type is of 4,745 tons displacement, and 347.7 feet long, with 3.53 inches instead of 3.94 inches armor, and carrying, in lieu of the six 6.2-inch guns, six quick-firing guns of 5.5 inches. These vessels, moreover, will have but 19 instead of 20 knots speed under forced draft. The *Pothau* will displace 5,245 tons, and, while carrying only the armament of the *Charner*, will have a speed equal to that of the *Dupuy de Lôme*. All the six vessels will be provided with torpedo ejectors. By the end of the current year the *Dupuy de Lôme* should be completed for active service; the *Chanzy* and *Latouche Treville* should be advanced 75 per cent.; the *Charner* should be advanced 70 per cent.; the *Bruix* should be advanced 33 per cent.; and the *Pothau* should be advanced 9 per cent. The six ships will cost, upon an average, $2,200,000 apiece before they can be made ready for sea. It is suggested that, at a very much smaller cost, it would be possible for us so to refit the *Achilles*, *Agincourt*, *Minotaur* and *Northumberland*, and perhaps also the *Black Prince*, *Warrior* and *Hector*, as to render them for all practical purposes even better than the new French vessels. The old five-masters are more like the new French ships than any of our large mod-

ern cruisers in the matter of proportions, for they are 6.73 times as long as they are broad; they could be given enormous coal-carrying capacity; they could be made to steam 18 or 19 knots with ease; they could easily bear twice the armament of the *Dupuy de Lôme;* and they already have armor which, though not of steel, is 30 per cent. thicker than that of the French vessel. Above all, they have magnificent gun-platforms.—"Marine Engineer."

Linois.—Monsieur Burdeau, the new French Minister of Marine, has just given the order to the Société des Forges et Chantiers de la Méditerranée for the construction in their shipyard at La Seyne, near Toulon, of a fast cruiser to be named the *Linois.*

The plans have been prepared by the director of the Society's La Seyne shipyard, Monsieur Lagane, late engineer constructor of the French Navy. The principal dimensions of the cruiser are: Length, 328 feet; beam, 34 feet 5 inches; depth, 22 feet 5 inches; displacement, 2,225 tons.

The propelling machinery consists of four independent sets of vertical triple-expansion engines, supplied with steam by six single-ended marine boilers. Twin screws will be fitted, and a speed of twenty knots, which it is expected will be exceeded, is demanded by the specification.

There is no battery, properly so called, the artillery being all on the decks. This artillery consists of four quick-firing guns of 14 centimeters (5.51 inch), mounted on sponsons at the sides of the upper deck; two quick-firing guns of 12 centimeters (4.72 inch), mounted, one forward on the forecastle, the other right aft. All six of these guns are protected by means of well-armored shields. In addition to the above, there are four 47-millimeter Hotchkiss quick-firing guns, situated on the bridges, and six Hotchkiss revolver machine guns of 37 millimeters, of which four are mounted on the military tops. There are also four torpedo-discharging tubes, one forward, one aft and one on each broadside.

The engines, boilers and other contents of the hold, are protected by means of a protective deck of steel, extending the entire length of the vessel, and beneath this again, in way of en-

gines and boilers, a splinter screen deck is fitted as an additional protection to these vital parts.

Two military masts, with double military tops, are to be fitted, and two chimneys are provided. The vessel will be lighted throughout by means of 140 incandescent electric lamps, whilst electric search-light projectors will be fitted on the bridges and in the tops. The necessary current of electricity is generated by Gramme dynamos, which are to be made of sufficient power to supply current to electric motors for the working of the larger guns, and the ammunition hoists of both these and the smaller pieces.

The *Linois* is designed for use as a scout to a fleet, and, as seen from the above list of artillery, is powerfully armed in addition; what, however, is regarded as its most important quality is its high speed, which permits the officer in command to accept or refuse combat, as he thinks fit, and in accepting, to have the advantage of choosing the conditions under which that combat should take place.

Shipbuilding Programme.—The voting of the supplementary budget for the Navy of the French Government has permitted the putting in hand of many vessels for which the plans have been for some time completed. These are the ironclad *Massena*, the cruiser *Linois*, and the torpedo-depot ship *Foudre*. The specifications and contract for the first-class cruiser *Pothau* have been submitted for competition to the principal private French shipbuilding companies, and as soon as the contract is concluded the vessel will be commenced. The Société des Forges et Chantiers de la Loire is about to be intrusted with the construction of a second-class cruiser, which will be named the *Descartes*, and which will have the following characteristics: Length, 305 feet; displacement, 3,620 tons; I.H.P., 9,000; speed, 18 knots per hour. The same company receives also for construction one of the three sea-going torpedo boats, *Flibustier*, *Ariel*, and *Tourmente*.

This budget also provides for the construction of the ten first-class torpedo boats which were included in the shipbuilding programme of the French Navy for the year 1892. These will receive the numbers from 182 to 191 inclusive. The ten boats are

of the same type as the boats numbered 145 and upwards, and will have their principal dimensions as follows: Length, 118 feet; displacement, nearly 80 tons; I.H.P., 1,000. They will be armed with two quick-firing 37-millimeter Hotchkiss guns.

For the above-mentioned ten boats a specified speed of 21 knots per hour had been decided on, but judging from the success obtained with regard to speed, on the steam trials, of the torpedo boats of the same type, numbers 147, 148 and 149, which have all given over 23 knots, the French authorities have thought it advisable to demand a speed of 23.5 knots per hour, being an increase of 2½ knots per hour on the speed originally anticipated for them.

The greater number of these boats are to be intrusted to Messrs. Normand, of Havre, for construction, who, by adopting the use of a boiler of the Thornycroft type, have obtained greater success as regards speed than any other French torpedo boat builders.

Great progress has been made within the last few years by the French contractors with reference to the speed of torpedo boats; and for the three sea-going boats laid down this year, namely, the *Chevalier*, *Corsaire* and *Mousquetaire*, speeds of 24½, 25½ and 24½ knots respectively are anticipated.

The *Wattignies*, constructed at Rochefort, recently went for her trial trip with successful results. The *Wattignies* is a vessel of 1,310 tons, and is fitted with two sets of triple-expansion horizontal engines constructed at the Creusot works. The cylinders of the engines are 23.5 inches, 35.4 inches and 55.6 inches in diameter respectively, by 35.4-inch stroke. Several trials were carried out, the results of which are shown below:

Duration of Trial.	H. P. Developed.	No. of Revolutions of Engine per Minute.	Speed Attained in Knots.
Six hours	2,785	124.2	16.7
Twenty-four hours	2,865	124.4	17.1
Four hours	4,012	135.2	18.06
Twenty-four hours	4,100	135.5	18.08

The torpedo scout *Levrier*, constructed at the arsenal of Lorient, recently completed a successful trial trip. The *Levrier*, together with her sister ship, *Léger*, is a vessel of 440 tons, and is fitted with two sets of triple-expansion engines, driving twin screws, constructed at the Creusot works. The cylinders of the engines are 15.1 inches, 24 inches and 36.1 inches diameter respectively, by 17.2-inch stroke. Several speed trials were carried out, the results of which are as follows:

Duration of Trial.	H. P. Developed.	No. of Revolutions of Engine per Minute.	Speed Attained in Knots.
Six hours	588	196.5	12.8
Twelve hours	1,525	276.6	16.9
Six hours	2,163	306.9	18.4
Three hours	2,238	310.8	18.4

GERMANY.

High Speed Torpedo Boats.—Herr F. Schichau, of Elbing, has attained an unprecedentedly high speed in one of several torpedo boats he is building for the Italian and Russian Governments, having got as a mean of one hour's continuous run in the open sea 27.4 knots. The dimensions are: Length, 152 feet 6 inches; beam, 17 feet 5 inches; displacement, 130 tons; bunker capacity, 40 tons. Each boat has two round-crowned locomotive boilers, which are protected by the coal bunkers. The working pressure is 13 atmospheres (105 pounds.) At the trials, the weights on board, in addition to 20 tons of coal were: Torpedo armament, 6 tons; guns, 2½ tons; crew (24 men), provisions, stores, and fire-arms, 4½ tons; drinking water 2½ tons; engine reserve parts and engineers' stores, 2 tons; ship's and boatswain's stores, 2½ tons—total, 40 tons. The guaranteed speed of 26.5 knots during a two hours' open-sea continuous trial was easily obtained, and during one hour's open-sea continuous run the speed was 27.4 knots. The number of revolutions was 325. The maximum thickness of the deck and hull plates of these boats is .226 inches. This is slightly diminished towards bow and stern.

The new British 200-ton boats ought, on account of their superior size, to be even faster.

Kaiserin Augusta.—On September 29th the *Kaiserin Augusta*, armored cruiser, built by the Germania Co., went on her trial trip. The trial was awaited with great interest by the German Admiralty and by the builders. The conditions for this trial, stipulated by contract, were rather severe, and, it is said, had never previously been fulfilled in such a manner at the first attempt. The vessel had to steam for eight consecvtive hours at sea, during which time the engines were to be worked up to their contract power, viz., 12,000 I.H.P., which they had to average for two hours. As a matter of fact, the trial trip lasted nearly 9 hours, and the mean of the I.H.P. during the fourth and fifth hours was 12,616, the maximum at one time being upwards of 13,000 I.H.P. The vessel has a displacement of 6,c00 tons and is fitted with three propellers worked by three independent sets of triple-expansion engines. The displacement on the trial trip was about 5,000 tons, and the average speed during the two hours full speed was 20.86 knots. The trial was made by the builder's own personnel, and it is believed that, when the vessel is in her service condition, with a properly trained crew of stokers, her average speed will be over 21 knots. Brief notices of this vessel have already appeared on pages 126 and 289 of the current volume of the JOURNAL.

Woerth.—On the 6th of August there was launched from the Germania Shipyard at Kiel a battle ship for the German Navy of 10,000 tons displacement.

A more successful launch has never been witnessed in Germany, and the interest was enhanced by the fact of the *Woerth* being the heaviest ship ever launched in that country. She had the whole of the armor belt in place, and weighed a little more than 5,000 tons.

The *Woerth* is one of the four battle ships which were voted by the Reichstag in 1889, and her principal dimensions are: Length, over all, 380 feet 6 inches; length between perpendiculars, 345 feet 3 inches; breadth moulded, 64 feet; depth of hold, 40 feet; draught, 24 feet; displacement, 10,000 tons.

The vessel has been built entirely of German materials, the steel castings and the armor plates having been supplied by Mr. F. Krupp, of Essen.

There is a complete armor belt of compound plates at the water line, 8 feet broad and 16 inches thick amidship, tapering to 12 inches at the ends, with an armor deck 2½ inches thick on the top from end to end and from side to side. The vessel is armed with six 11-inch Krupp breech-loading guns, six 4¼-inch and ten 3½ inch quick-firing guns. The six big guns are placed in barbettes, which are protected by 12-inch compound armor, and the auxiliary armament is also protected by thinner armor.

The propelling machinery consists of two sets of vertical triple-expansion engines, which will indicate 9,000 H.P., steam being supplied by twelve single-ended boilers placed in four separate rooms. The speed will be 16 knots. The whole vessel is lighted by electricity, and provision against torpedo attack and against damage has been made by minute subdivision of the double bottom and of the spaces above the double bottom.

The vessel was designed by the German Admiralty, but the engines and boilers for all four vessels have been built after the designs of the Germania Company, and the engines and boilers of the second-class torpedo boats, of which each vessel carries one, have even been supplied by the Germania Company, their superiority over other designs having thus been freely acknowledged by the German Admiralty.

Hildebrand.—On the same day, but an hour earlier, a smaller armorclad was launched from the Imperial Dockyard, also at Kiel. This is one of the ten coast defence armorclads of 3,600 tons, the first of which was built two years ago by the Germania Company; two others are building at Bremen, one at Wilhelmshaven, two at Kiel, one at Danzig and the remaining ones will be proceeded with next year.

MERCHANT STEAMERS.

New Steamer for Fall River Line.—The new passenger and freight steamer (not yet named) for the Old Colony Steamboat Co. (Fall River Line, New York to Boston) will be built, in general, after the style of the famous *Puritan* of that line, the only marked changes being an increase in length, the style of engines and boilers, and placing the dining room on the main deck instead of in the hold.

The new boat will be made as long as can be accommodated safely in the New York dry docks.

The change in style of engine from the compound-beam to compound double-inclined engine is made to increase the stability of the vessel, for, while the *Puritan* cannot be excelled as a comfortable sea boat, it was found that to give the new boat the increased power desired would make the weight of beam and upper part of engine greater than the increase in stability of new hull would safely permit. The double-inclined engine has other advantages for a vessel of this size. It has practically no centers, and the use of an enormous low-pressure cylinder is avoided.

The change in boilers is made to give a greater range of working pressure.

The change in the position of the dining room is one of the improvements always sought by the management of the line for the pleasure and comfort of its patrons. The boat will be completed in 1894.

We give below the general dimensions and other particulars.

Contractors and engine builders, W. & A. Fletcher Co., Hoboken, N. J. Hull builders, The Delaware River Iron Ship Building and Engine Works, Chester, Pa. Joiner, William Rowland, New York.

Hull.—Length on L.W.L., 423.5 feet; length over all, 440.5 feet; beam of hull, 52½ feet; beam over guards, 92½ feet; mean

draught, light, 12 feet; displacement, light, 4,550 tons; cargo capacity, 800 tons; passenger and crew capacity, 1,400.

Engine.—Type, compound double-inclined; number of cylinders, 4; two H.P. cylinders, 51 inches diameter; two L.P. cylinders, 95 inches diameter; stroke, 132 inches; revolutions for full power, 25; indicated horse power, 8,000; air pumps, type, connected; circulating pumps, type, independent; condenser, type, surface; condenser, cooling surface, 16,000 square feet.

Wheels.—Feathering wheels, diameter over buckets, 35 feet; width of buckets, 14 feet; depth of buckets, 5 feet; immersion of buckets, light, 7 feet.

Boilers.—Number of boilers, 10; type of boilers, Scotch single-ended; mean diameter, 14 feet; length of boilers, 14 feet 6 inches; number of furnaces in all, 30; type of furnaces, corrugated; inside diameter of furnaces, 44 inches; total grate surface, 850 square feet; total heating surface, 24,000 square feet; built for working pressure of steam, 150 pounds per square inch; arranged for natural or forced draft, as may be desired.

The following is a comparison of the dimensions of the recent steamers of this line in the order of their inception:

	Pilgrim.	*Puritan.*	*Plymouth.*	*Mayflower.*
	ft. in.	*ft. in.*	*ft. in.*	*ft. in.*
Length on water line	372 5	403 4	351 8	423 6
" over all.....	387 0	419 4	367 0	440 6
Beam molded.....	50 0	52 0	50 0	52 3
" over guards.....	87 3	91 2	87 0	92 0
Depth molded lowest part.	17 6	20 6	21 0	20 6

The *Mayflower* (the name of this vessel is not definitely settled), like the latest additions to this line, with the exception of the *Pilgrim*, is to be built of steel, and it is the intention on the part of both owners and builders to make the present proposed addition to the line the greatest success in point of fineness of lines, power, speed, capacity and elegance yet obtainable, to accomplish which, when compared with the palatial steamer *Puritan*, is no mean task.

Like the three first-named boats, the *Mayflower* will be non-

sinkable, non-burnable, and, in a word, non-destructible. She is built on the double-hull, longitudinal, cellular system, with a length of double bottom of 340 feet, having in all fifty-two compartments, besides which, the hull is subdivided above the double hull and at ends of vessel by means of bulkheads and flats into nine additional water-tight spaces, making in all sixty-one. The freeboard of this monster is sufficient to float her with any ten of the compartments within the double bottom full of water, or with any two of the compartments above the double bottom filled.

As an instance of the non-sinkable condition of these vessels, two examples can be cited appropos of the present description.

The first case happened to the *Pilgrim*. While coming down East River on a remarkably low tide, she struck a rock which tore through the outer hull for a length of 85 feet by a width of 4 feet. The inner hull, being uninjured, enabled the *Pilgrim* to proceed to her dock and discharge her passengers and freight.

The second accident of this character happened to the steamer *Puritan* in Hell Gate, where, in trying to keep out of the way of other boats, she was forced over too far and tore her bilge against the rocks for a length of 120 feet by a width of 5 feet. After discharging passengers and freight, she proceeded to the dry dock, where, after lying for forty-eight hours and being hauled out, not a drop of water was found to have entered the inner hull.

As a synopsis of the construction of the present addition to the above fleet could be mentioned the following: The center keelson or backbone consists of a steel plate 42 inches deep, connected to the double plate keel by two strong angle bars, and to the floor stringer plates, also double, in the same manner. On each side of this keelson are five longitudinals extending the whole length of the vessel, the third one from center being water-tight. These longitudinals vary in depth from 42 inches to 36 inches as their position approaches the sides of the boat. The margin plate bounding the outer edge of the water bottom is 24 inches wide, and is made water-tight by stapling around the frames, the reverse bars being cut at this point. Above the water bottom the frame of the vessel consists simply of the two angles, frame and reverse bar, riveted together. Midway between the

margin plate and deck stringer is a side stringer 27 inches wide, supported by gussets on alternate frames.

The gunwale structure of the vessel consists of a lodger plate secured to frames by gussets, and upon which the main deck beams (of I section) are landed. This lodger plate is the depth of the beams (which amidships are 12 inches deep) below the deck stringer proper, this latter being a string plate extending over the gunwale on the outside to take the guard beams, and extending inboard on the beams to take the deck. The proportion of stringer on the deck to that overhanging the gunwale is as 3 to 1.

The transverse frames are thirty inches, center to center, strongly riveted to the center keelson and to the longitudinals (which pass through them), and forty-two inches deep within the double bottom amidship. They are made up of the bracket and skeleton construction alternating. At distances of about thirty feet, water-tight floors are inserted as boundaries for the water-tight compartments. Directly under the engine keelson, all the floors are made on the bracket order with more metal left in them, top and bottom, than generally, throughout the double hull, to insure absolute freedom from motion in the bottom when engines are working.

As already stated, the deck beams of this vessel are of I section, and are not, as is customarily done, riveted directly to the frame by welded knees. As the riveting of these beams to alternate frames would make too great a shift, the builders have adopted the shelf or lodger-plate construction. The shelf, being strongly secured to every frame by gusset plates, makes a good and substantial support for the beam end, and as they are 45 inches apart, every other one comes over and is riveted to a frame which extends through and above the lodger plate. Besides, as the ends of the beams come between the lodger and deck-stringer plates, they will be securely riveted to both through top and bottom flanges.

There are other features in the construction of this boat which show a judicious use of material, enabling the greatest strength to be obtained on the least practicable weight.

The cargo capacity is eight hundred tons (800 tons), which is carried on the main deck forward of the engine space. The draught, when loaded, is expected to be about 13 feet, and mean displacement 5,200 tons.

The joiner work and decorations will display the highest degree of excellence, surpassing, if possible, the *Puritan* in this respect. There will be three hundred and fifty or more rooms for passengers, besides those for the President, Captain, Purser and other officers, as well as the usual quarters for waiters, crew and other attachés of the boat.

In outward appearance the *Mayflower* will resemble the *Puritan*, having two steel masts and two smoke pipes. As there is no walking beam to be taken care of in this boat, as in the *Puritan*, there may be a little difference in her appearance about the dome deck, approaching in this respect nearer to the *Plymouth*.

At some future day it may be possible to give a much fuller description of this vessel, as well as comparative descriptions of the other boats now in commission.

[For the foregoing excellent descriptions of the hull and machinery of this splendid steamer, the readers of the JOURNAL are indebted to Stevenson Taylor, Esq., Vice-President and General Manager of the A. & W. Fletcher Co., and R. T. Dickinson, Esq., Superintending Naval Architect of the Delaware River Iron Shipbuilding Co.—EDITOR.]

Campania.—The Fairfield Shipbuilding and Engineering Co., Limited, of Govan, Glasgow, launched on August 8th, from their yard, the twin-screw steamer *Campania*, which has been built for the Cunard Co., and which will be one of the two largest and finest vessels afloat. The ship is of the following dimensions: Length, over all, 620 feet; breadth, extreme, 65 feet 3 inches; depth to upper deck 43 feet, and gross tonnage about 12,500 tons. The vessel has a straight stem and elliptic stern, topgallant forecastle and poop, with close bulwarks fore and aft, the erections above the upper deck consisting of two tiers of deck-houses, surmounted respectively by the promenade and shade

decks. When completed the vessel will have a fore-and-aft rig, with two pole masts.

With regard to the gigantic proportions of the ship, it will readily be understood that the greatest care has been necessary, both in the method and details of construction, in order that the requisite strength might be kept up throughout the entire structure. All scantlings have been specially arranged, and every advantage taken of the improved sections of steel, known as Channel and Z bars, &c., in order to increase the strength without adding unnecessarily to the vessel's weight. The vessel has undergone special survey while building, and will take her place in the highest class at Lloyd's. The bulkheads, sixteen in number, have been fitted somewhat in excess of the recommendations of the recent Board of Trade Committee on the fitting and spacing of watertight bulkheads, and will enable the vessel to float with any two, or in some cases three, compartments open to the sea. In addition to these qualifications, the vessel has been built to meet Admiralty requirements for serving as an armed cruiser in times of war.

The decks have been specially arranged and strengthened to carry guns. and the steering gear fitted of a special type under water. Arrangements have also been made for the suitable protection of the vital parts of the ship. The keel is of the flat plate description, and a cellular double bottom extends fore and aft, and is fitted with draining and flooding arrangements to make it available for the purposes of water ballast. The steel plating, which is in lengths of 26 feet and over, is fitted on the lap-butt principle, from the keel-plate up to about the water-line. Above that the plates are butted and fitted with double straps. One of Wasteneys Smith's patent stockless anchors, weighing nearly 6½ tons, will be carried.

The construction of the vessel at the stern is somewhat peculiar. Although fitted with twin-screws, there is an aperture in the stern frame similar to that in a single-screw steamer. This is provided that the propellers may work freely, though they are fitted close to the center line of the ship in order to prevent damage to or from the quay walls. No struts are fitted at the

stern frame to support the outer end of the shafts as in most other twin-screw steamers. These are in the present case dispensed with, and the frames of the hull are bossed out and plated over right aft, so as to form the stern tubes. At the outer ends of these, strong castings of steel are fitted. These serve the purpose of shaft brackets, and being in continuation of the lines of the hull are calulated to offer the least resistance to free propulsion.

The rudder, which is fitted entirely under water, is of the single plate type, being formed of a heavy steel casting with massive arms, between which a thick plate is fitted and riveted. The whole arrangement, therefore, of the stern of the *Campania* impresses one as being uncommon in other respects than that merely of immense size.

Besides the superstructure already mentioned there are four complete tiers of beams, all of which are plated over with steel, and sheathed with wood planks, forming what are termed the upper, main, lower and orlop decks. The last is used for cargo and refrigerating chambers, store rooms, &c. The other decks are entirely devoted to the accommodation of passengers, with the usual dining and social saloons. Sanitary arrangements are also provided on the various decks where found most convenient.

All the latest improvements are being introduced, and no expense spared on anything calculated to render traveling at sea more comfortable and enjoyable. The casings around the boiler-rooms are double, the intervening space being filled with a material which is at once a non-conductor of heat and sound. The *Campania* will be thoroughly ventilated by both natural and artificial means. The greater number of the sidelights are fitted with an arrangement for the free admission of air, even when the lights are closed during rough weather. A complete system of steam heating pipes will provide for the comfortable warming of all the living spaces. A very complete installation of electric light on the most extensive scale is also provided.

The *Campania* will have two sets of the most powerful triple-expansion engines that have yet been constructed, fitted in two separate engine-rooms, there being a dividing center-line bulk-

head between them, fitted with watertight doors for the necessary purposes of communication. Each set of engines consists of five inverted cylinders, viz., two high pressure, one intermediate pressure, and two low pressure cylinders, the two high pressure cylinders being placed above the low pressure ones. These are arranged to work on three cranks, set at an angle of 120° with each other, and all having the same stroke. The high pressure cylinders are each fitted with a piston valve, the intermediate and low pressure ones with double piston valves, all of which are worked by the usual double eccentrics and Stephenson double-bar links. The reversing engines are of the steam and hydraulic direct acting type, made by Messrs. Brown Bros. & Co., Edinburgh. The reversing engines are also fitted with their patent automatic emergency gear. The crank shafts are all built up and are interchangeable, and, with the tunnel and propeller shafting, are of mild steel, forged by Messrs. Vickers, Sons & Co., Sheffield. The condensing water is circulated through the condensers by four large centrifugal pumps, each driven by independent compound engines. These pumps may also, if required, pump water from the bilge in case of damage to the hull. An elaborate system of piping is fitted throughout the ship, and connected to the various auxiliary pumps for filling and emptying the ballast tanks, pumping out bilges, pumping water on deck in case of fire, and other purposes.

In the engine room are also fitted four evaporators to produce the necessary fresh water to make up the loss incurred through working. There is also a large feed-water heater, two auxiliary condensers with pumps, and all the most modern appliances to economize fuel and labor. Steam for the main engines is generated in twelve large double-ended boilers. There is also a large single-ended boiler for supplying steam for the electric light, refrigerating and other auxiliary machinery. In addition, a small single-ended boiler will be fitted on the lower deck for supplying steam to the distilling condensers, heating pipes, &c. The main boilers are arranged in two groups, having one funnel for each group.

The launch of the *Campania* was noteworthy, as she is

computed to measure when completed some 12,000 or 12,500 tons.

Viewed as an undertaking in connection with a branch of the naval architect's and ship-builder's work and responsibilities, which have grown more and more arduous, responsible and difficult as the dimensions and weight of vessels have increased, the successful launch of the *Campania* was a triumph in the highest sense of the word for all concerned, and for the Clyde as a ship-building district. The weight of the *Campania*, as she sat on the ways ready to take the "plunge," was, as near as could be computed, 9,000 tons. The weight of the next heaviest ship ever launched on the Clyde—H.M.S. *Ramillies*, in March, last—was about 7,000 tons, while that of the heaviest battle ship ever removed from a Government slipway was that of the *Repulse*, launched from Pembroke Dockyard, also in March last, amounting to 7,650 tons. The *Campania's* transference to her "native element" was the occasion of, perhaps, the largest concourse of spectators ever brought together on the banks of the Clyde, and never, perhaps, in the whole history of shipbuilding on that river have hearty and merited congratulations on the success of a launch been more freely showered upon the chief performer in such a piece of work than were bestowed upon Mr. R. Saxon White, the general manager at Fairfield. The sister ship to the *Campania*, which is now wholly in frame and partially plated, will be launched early in November, which in some respects will be even a more arduous undertaking. She is to be called the *Lucania*.

Gigantic.—The credit of building the largest and fastest steamers afloat has not remained very long with the Cunard Company, if newspaper reports are credible. Although definite information has not yet appeared as to size and speed, it is well known that the White Star S. S. Company have placed an order with Harland & Wolff, of Belfast, who have built all their ships, for a steamer to be called the *Gigantic*, and which is expected to surpass the *Campania* and *Lucania*, of the Cunard Line.

The dimensions of this vessel are said to be 700 feet long (over all) by 65 feet 7½ inches beam, and the horse-power is to

be 45,000, driving triple-screws as in the *Columbia* of our Navy. The sustained sea speed is expected to be about 22 knots.

It may not be without interest to give some estimates as to size and speed of this vessel in advance of the knowledge of the actual figures. As the White Star ships have been of quite uniform proportions, it may be fair to assume that the same will hold true in this case. This would make the length on water-line about 650 feet. The draught of vessels entering New York harbor cannot exceed 26.5 feet, and all the large vessels have that draught. Taking these figures, and allowing a co-efficient of fineness of about 62 per cent., we arrive at a displacement of 20,000 tons.

Taking the progressive trial data of the *Yorktown*, and bringing them to 20,000 tons, would give 23 knots speed for 45,000 I.H.P. This agrees very well with the claimed value of 22 knots as the sustained sea speed.

Richard Peck.—This fine vessel, which just now holds the record as the fastest steamer on Long Island Sound, was built last year for the New York and New Haven S. S. Co., by the Harlan & Hollingsworth Co., of Wilmington, Delaware. She represents a departure in Sound practice, as she has twin screws, and is the first vessel of this type built for the Sound.

Length over all, 315 feet; length on water line, 300 feet; moulded beam, 48 feet; mean draught, 10 feet 6 inches; displacement, 2,500 short tons or 2,230 tons of 2,240 pounds. The engines are of the usual vertical type and triple-expansion, with cylinders 24, 38 and 60 inches diameter, by 30 inches stroke. There are six single-ended cylindrical boilers, 13 feet in diameter and 11 feet 6 inches long, with an aggregate grate surface of 390 square feet, and heating surface 11,670 square feet. The propellers are 10 feet 6 inches diameter and 15 feet 6 inches pitch; they are of cast iron.

Two runs over the measured statute mile in Delaware bay (described on page 132 of the current volume) were made with the following results:

	Speed.	I.H.P.	Revs.	Steam.
With tide, . .	21.052	3,259	120	150
Against tide, .	18.000	3,914.6	125	150

Mean speed, 19.526 statute miles or 16.96 knots. The displacement was 2,280 short or 2,040 long tons.

On a later trial over the measured mile, a mean speed of 20.25 statute miles was obtained, but owing to an accident to one of the indicators, the horse power can not be given.

The foregoing information has been kindly furnished by T. Jackson Shaw, Esq., Superintending Engineer of the H. & H. Company.

El Rio.—This fine steamer was launched at Newport News October 26th, and is an exact duplicate of *El Norte* (described in the last number of the JOURNAL), except that she is built of steel, and has only two masts instead of four. Like her sister ship, she is the design of Horace See, Esq., the Consulting Engineer of the Newport News Shipbuilding and Dry Dock Co., who has kindly furnished the following data:

Length over all, 406 feet; length on L.W.L., 390 feet 3 inches; beam moulded, 48 feet; depth moulded, 33 feet 9 inches; mean draught, 23 feet; displacement, 7,500 tons; cargo capacity, 4,250 tons.

There is a vertical, triple-expansion engine driving a single screw, with cylinders 32, 52 and 84 inches diameter, by 54 inches stroke. The I.H.P. is about 3,800 at 79 revolutions per minute. The air pump is worked from the main engines, while the circulating pump is centrifugal and driven by an independent engine.

There are three double-ended steel boilers, 20 feet 6 inches long and 13 feet 10 inches diameter, each having six corrugated furnaces, with an internal diameter of 44 inches. The steam pressure is 165 pounds.

It is expected that *El Rio* will be completed early next year.

The accompanying cards from the engines of *El Norte* are interesting as showing what may be expected of these vessels. *El Norte* has made an average speed between New York and New Orleans of 16.25 knots.

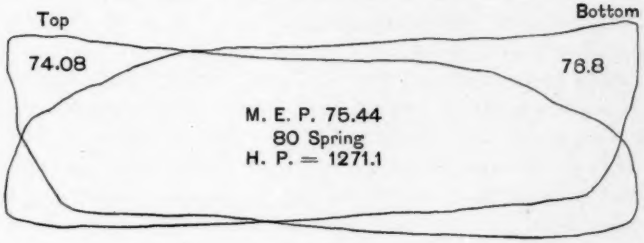
Alabama.—The largest vessel of the Bay Line, the passenger propeller *Alabama*, was launched on October 1st at the yard of the Maryland Steel Company, Sparrow's Point, near Baltimore. The *Alabama* is 305 feet long over all, 18 feet 1 inch moulded depth,

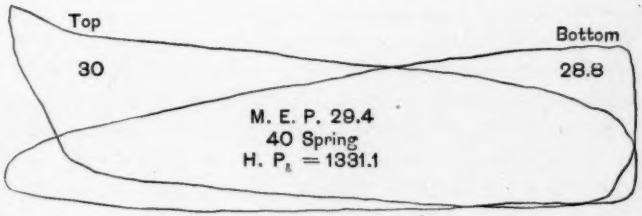
Cards #6 taken at Sea, from, S.S. "El Norte."

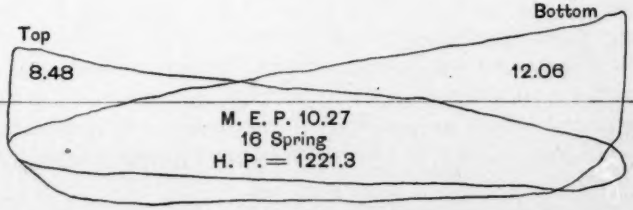
10 A.M. Sept. 28th. 1892.

Cylinders 32." 52" & 84" by 54" Stroke. Piston Rod 7" dia.

Boiler Gauge, 162 lbs. Vacuum, 25 lbs. Revolutions per minute, 79.







Total I. H. P. = 3823.5

40 feet beam at the water line, 43 feet beam over the hull at deck and 55 feet over the guards. In giving out this contract the Baltimore Steam Packet Company expressed their desire to have a heavily built boat, in which every precaution should be taken to reduce vibration both in the scantling of the hull and design of the engine. The boat is, therefore, built in steel fully up to iron sizes, and the utmost pains have been taken to design the engine foundations and general framing of the boat to carry out their views. The keel is of the center through-plate construction, a type seldom used on account of its great expense, but the most admirable way of combining material for the purpose ever employed. The frames are 4 inches by 3 inches by 10 pounds doubled under engines and boilers; the reverse frames 3 inches by 3 inches by 7¼ pounds, running to the main deck and to the lower deck alternately, and doubled under the engines and boilers. The frames in the wake of the engines are all web frames 12 inches deep. The outside plating is 22½ pounds, the sheer strake is 27½ pounds and garboard 24 pounds. Doubling is worked forward on the water line to resist ice.

The machinery consists of a four-cylinder triple-expansion engine having two low-pressure cylinders. The cylinders are 24½ inches, 40 inches, and two of 47 inches by 42 inches stroke. They rest on eight cast-iron housings of the usual design. Pursuing the leading idea of avoiding vibration, the reciprocating parts have been kept as light and the fixed parts as heavy as seemed consistent. To the end first mentioned steel pistons have been employed. The cranks are arranged so that two pistons of approximate weights shall always be opposite and ascending and descending forces balanced. There is an independent condenser with centrifugal circulating pump, but the air pump is worked off the main engines. The high-pressure cylinder is provided with piston valves. All the others are slide valves. There will be steam-reversing and steam-turning gear. A Davidson compound pump is used for the independent feed pump. The propeller is 12 feet 9 inches diameter, built up with cast-steel blades. Steam of 160 pounds pressure is supplied by four boilers aggregating 264 square feet of grate. They are 12

feet 9 inches diameter and 10 feet 8 inches long inside, with three 44-inch Purves furnaces in each boiler.

In joiner work the *Alabama* is expected to be much the finest vessel of the line. On the lower deck she will have quarters for the officers and crew forward, and for second-class passengers; also a cargo hold. Aft of the machinery on this deck she will have kitchens, pantries, waiters' quarters and a beautiful dining-saloon finished in red oak with buffets along the sides much like some of the Sound steamers. The main deck forward and in the wake of the machinery spaces will be given up for freight, while aloft it will be occupied by the social hall and ladies' cabin. The former will be finished in red oak in a novel and handsome design, the latter in oak and mahogany. The saloon deck will have its ranges of staterooms carried right forward and aft. The finish will be of white and gold in a unique design with pilasters of mahogany. The furnishings will be by P. Hansen Hiss & Co., of Baltimore, and the electric light fixtures by Dekosenko, Hetherington & Co., of Philadelphia. Forward there will be a large well opening up through the gallery deck; very handsome staircases in mahogany will lead to the gallery deck, which contains about 22 staterooms. The joiner work of the gallery deck will also be finished in white and gold with mahogany trimmings. The electric light plant is by the Edison General Electric Co., and provides for 360 incandescent lamps and a searchlight of 6,000 c. p.; also electric fans for the ventilation of the dining saloon. Altogether the *Alabama* is intended to be the finest and fastest steamer which plies on Chesapeake Bay.—" Seaboard."

Alberta.—On September 8th the S. S. *Alberta*, built by Messrs. Wm. Gray & Co., of Hartlepool, England, went on her trial trip, being completed and ready for her first voyage. She is a fine example of the shipbuilding and engineering skill of the firm, and is built to take Lloyd's highest class. She is 400 feet long over all, 40 feet 6 inches in breadth, and 31 feet 6 inches deep. The web-frame system has been adopted in her construction, and there is a cellular double bottom for water ballast. There are exceptional facilities provided for the rapid and efficient working and haulings of the ship and her cargo, the poop aft being also

provided with patent direct steam capstan windlasses by Emerson, Walker & Co., for the safer manipulation of so large a ship. The ship is also fitted with patent stockless anchors, stowed away in the hawse pipes, and Harrison's steam steering gear, which is made to manipulate Hastie's system of tiller gear by means of a special safety apparatus, which allows of a considerable amount of flexible movement of the rudder when the same may be struck by the sea. All of the deck steam engines, which are a large number for a cargo boat, are provided with steam from the two large upright multitubular donkey-boilers. The main engines for propelling the vessel are supplied from the Central Marine Engine Works of Sir William Gray & Co., the cylinders being 27½ inches, 43½ inches and 73 inches in diameter respectively, with a piston stroke of 46 inches. These fine engines are supplied with steam from two large double-ended boilers, working at a pressure of 160 pounds per square inch. The engines are among the largest that have hitherto been supplied from these comparatively new engine works, and are the 132d set built there. They are of the type now universally adopted at the Central Engine Works, having the high-pressure engine in the middle, the cylinders being supported at the front by three large wrought-iron double-collared columns, making the working parts of the engine thoroughly accessible.

The design of these engines throughout is the same as that of those which passed so successfully through the searching trials carried out by Professor Kennedy and his staff on board the S. S. *Iona*, and others like the *Rangatira*, *Tekoa*, *Calliope*, *Heighington*, *Ariel*, *Inchbarra*, and many more, which throughout long voyages and severe tests have kept up the reputation of these builders, both for high economy in consumption and for that good workmanship which results in long voyages being made without a stop, and in long periods being passed in full work with the minimum of wear on the working parts. These good qualities, therefore, are possessed also by the machinery of the *Alberta*, and indeed she possesses also the latest touches of improvement devised by the builders. Her boilers, also, although double-ended, are made on the plan so strongly advocated at the

Central Engine Works, inasmuch as the shell plating is welded and flanged and fitted with flat end plates. The advantage of this method is again illustrated on board the *Alberta* by the fact that even on this first steaming there was not the slightest sign of leakage to be found about the boilers at any part. It is, however, after months and years of working that this system is most conspicuous in its results, as example after example can be now pointed to in which the boilers have remained practically quite tight and free from those destructive results of leakage to which the ordinary type of boiler is so subject.

The vessel went on her trial trip without any cargo or coal in bunkers, the water tanks alone being filled, so that the trial represented simply a test of the good running qualities of the machinery, and not one of the capabilities of the ship in the way of speed, as the propeller had not half of its diameter in the water. Under these unfavorable circumstances, however, the vessel made 10½ knots at about 65 revolutions, and 12½ knots at about 78 revolutions. The important matter, however, was that the engines ran for hours at these high speeds of revolution without any trouble whatever, all the bearings keeping quite cool, and not requiring the application of water or other special attention.

The engine room of this vessel is of large capacity and well ventilated, and is so well finished in every way, and so replete with those conveniences that conduce so much to the efficient care expended upon the machinery by those in charge, as to delight the heart of a superintendent or seagoing engineer. An exceptional amount of care has been expended on matters which are ordinarily supposed of but minor importance, such, for instance, as the provision of an unusual number of gratings, platforms, ladders and hand railings, so that there is no point to which an engineer requires to devote his attention that is not perfectly accessible in the easiest and most comfortable manner. In such matters also as the pipe arrangement, a new departure has been made at the instance of Mr. F. J. Pilcher, superintendent engineer, all tank and bilge pipes being brought, where possible, above the level of the starting platform, so as to be well within reach for painting and repairs. The bilge suction boxes,

too, are of special design, being distributed at various points in the engine room, and arranged so that they can be cleared from obstructions without it being necessary for a man to descend into the bilges for that purpose. The propeller, also, is unusual for a cargo boat, inasmuch as it is made of Stone's bronze instead of the ordinary cast iron, but in point of design it contains those special features which characterize all the propellers made at the Central Engine Works, and the casting of the *Alberta's* propeller was carried out at these works, a furnace having recently been erected there for the purpose of melting large quantities of bronze.

Australia.—On July 19th between three and four o'clock in the morning, Messrs. Caird & Co., shipbuilders, Greenock, Scotland, launched for the P. & O. Co. the magnificent 7,000-ton steamer *Australia*—a sister ship to the *Himalaya*, these two steamers being the largest ever built in Greenock. It was considered advisable to launch the ship at this unusual hour owing to the fact that the tides during the day were so low, and it was found there was considerably more water at the end of the ways during the night. The launch was a highly successful one. Her dimensions are: Length, 466 feet; breadth, 52 feet; depth, 37½ feet; and of 7,000 tons gross. She will be supplied by the builders with triple-expansion engines indicating 10,000 H.P. (at sea), and capable of driving her at 18 knots, or over 20½ miles per hour. She will be furnished with three double-ended and three single-ended boilers, and twenty-seven furnaces, fitted with Howden's patent system of forced draft.

The *Australia*, which has been built of Siemen's best steel, and to Lloyd's highest class, has a full poop and topgallant forecastle, with long bridge-house amidships. She will be smartly rigged with four pole masts, carrying fore and aft canvas. She has four decks, and splendid accommodation for first and second saloon passengers. The main saloon is 60 feet long by the whole width of the ship. The ship is lighted by the electric light throughout by Messrs. Siemens Brothers, London; the electric bells are by Messrs. Zimdar & Co., London; the refrigerating machinery by Messrs. Haslam & Co., Derby; the steam

steering gear by Messrs. Muir & Caldwell; the hydraulic cranes and machinery of a similar kind by Messrs. Armstrong, Mitchell & Co., Liverpool. Every attention has been paid to the convenience of passengers in the matter of baths, lavatories, &c. Special features in this connection are the spray rooms for both first and second saloon passengers, who will no doubt highly appreciate these in hot latitudes. After the launch, the *Australia* was towed to the Victoria Harbor, where her engines and machinery will be put on board.

Gedeh.—On August 24th there was launched, from the yard of the Royal Shipbuilding & Engineering Co., at Flushing, a steel screw steamer, built on account of the Rotterdam Lloyd Line of mail steamers to Batavia, and of the following dimensions:—Length, 348 feet; breadth, 38 feet; depth, moulded, 27 feet; built to the highest class of Veritas. This vessel has accommodation for 39 first-class, 24 second-class, and 300 third-class passengers. The engines are to be supplied by the same builders, and are quadruple-expansion, with cylinders 24 inches, 36 inches, 48 inches and 72 inches by 42 inches stroke. The working pressure is 200 pounds. They are calculated to indicate 2,000 H. P. at sea, and to drive the vessel 12 knots. This is the fifth set of engines of this type which the Rotterdam Lloyd Co. have running in their line, the coal consumption of which is exceedingly low.

An Economical Steamer.—As illustrating the performance of a first-class modern cargo steamer, it is interesting to note some particulars of a voyage just completed from the Tyne to Madras by the S. S. *Heighington*, a vessel belonging to the Hudson Shipping Co., of West Hartlepool, built by Wm. Gray & Co., and engined at the Central Engine Works. The vessel left the Tyne fully loaded, and with a total displacement of 6,100 tons, on the first day of July last, having on board a deadweight of cargo and bunkers of about 4,300 tons. She arrived at Madras on the 6th August, having run 7,512 knots in 830 hours, which is an average of rather over 9 knots per hour for the whole voyage (deducting for the Suez Canal), on a coal consumption of 14.3 tons per day for all purposes, or about 13 tons per day for propulsion only.

When so large a vessel as the *Heighington* can be driven at an average speed of nine knots per hour on so small a consumption as named above, and which represents the carrying of one ton of deadweight one mile for an expendlture of half an ounce of coal, there can be no wonder at the older class of steamers having to be laid up whenever freights are at all depressed. There appears, therefore, to be only one way out of the difficulty for the shipowner, and that is to provide his vessels with the most economical machinery, bringing them up to present day performance. It is worth noting that, amongst other improvements possessed by the *Heighington*, is a set of Mudd & Airey's patent metallic piston-rod packing, which, after a year's experimenting, has been brought to a high state of perfection, and is said to greatly aid the economy of the engines by reducing friction of the stuffing boxes to such an extent as to allow the engines to run several revolutions faster in the same gear. Such an improvement may seem small in itself, but the shipowner cannot afford in these times to everlook anything tending to increase the speed of his ship or reduce his coal bill.

Chicora.—Trials were made on the steamer's regular trips on Lake Michigan, between Chicago and St. Joseph, a distance of 60 miles. The first trial was for the purpose of ascertaining the maximum power of the machinery. A strong head wind was blowing with no sea on. Steam was kept up to 160 pounds, with full throttle. Cards were taken every hour throughout the trial. Steam, vacuum, revolutions, air pressure and temperature of chimney were noted with each set of rods taken.

The engine was kept at full speed for about one and one-fourth hours. No difficulty was experienced in keeping full pressure of steam in the boilers with an air pressure of 1⅝ inches at the fan, and ⅝ inch in the ash pit.

It was found that, in order to reduce the speed to about 94 revolutions per minute without throttling, the H.P. valve had to be linked up to full extent of arm, and at 88 revolutions all three valves had to be linked up.

The second trial was made for the purpose of finding the best

point of cut-off, to obtain as nearly as possible an equal distribution of power in the cylinders.

The best results were obtained by working the intermediate cylinder in full gear, and linking up the L.P. one inch from full gear.

As no means were at hand for weighing or measuring the coal, no account was taken of the amount used during trials.

Vessel.—Length over all, 209 feet; length of keel, 196 feet 4 inches; breadth, 34 feet 8 inches; depth, 15 feet 3 inches; mean draught on trial, 9 feet 8 inches; displacement on trial, 1,240 tons.

Engine.—Cylinder diameters, 20, 33 and 44 inches; cylinder stroke, 42 inches; propeller diameter, 11 feet; propeller pitch, 17 feet.

Boilers.—Number, 2; diameter, 12 feet; length, 11 feet 6 inches; number of furnaces each boiler, 2; diameter of furnaces, 3 feet 7 inches; grate area, 82.32 square feet; heating surface, 3,162.64 square feet; ratio heating to grate, 37.7 to 1; Howden's system of forced draft; diameter of fan, 48 inches; area of air main, 576 square inches; size of fan engine 7 × 7 inches.

PERFORMANCE.

	Maximum.	94 Revs.	88 Revs.
Boiler pressure, pounds	160.00	160.00	160 00
Vacuum, inches	21.00	21.5	21.5
Revolutions per minute	102.00	94.00	88.00
Indicated horse power, H.P. Cylinder	292.69	299.73	230.77
" " Int. Cylinder	563.66	444.77	317.50
" " L.P. Cylinder	648.63	453.45	393.73
" " Total	1,504.98	1,197.95	942.00
Revolutions of fan per minute	470	470	470
" " engine per minute	220	220	220
Temperature of chimney, Fahr.	630°	.....	580°
Heating surface per I.H.P. sq. ft	2.10	2.64	3.35
I.H.P. per sq. ft. of grate	18.16	14.55	11.44
Air pressure, ash pit, inches water	.625	.625	.625
" " main, inches water	1.625	1.875	1.875
Distance run, miles	.....	60	.....
Running time, hours	.....	4	.....
Speed of vessel, miles per hour	.....	15	.....
Slip of propeller, per cent	.....	17.3	.....

Result obtained by working intermediate full and linking L.P. up to 1 inch from full gear: Boiler pressure, pounds, 160; vacuum, inches, 21.5; revolutions per minute, 96; indicated horse power, H.P. cylinder, 324.67; Int. cylinder, 437.88; L.P. cylinder, 480.35. Total, 1,242.90.

The *Chicora* was built by the Detroit Drydock Co.—"Seaboard."

Samoa, the Largest Cargo Vessel in the World.—The steamer *Samoa*, the largest vessel yet built on the Wear, and said to be the largest cargo vessel in the world, was successfully launched from the yard of W. Doxford & Sons, Pallion, England, on October 22d. The vessel, which is built of steel, is 465 feet long, 52 feet breadth, and 36 feet depth, has a gross register of 6,400 tons, deadweight capacity of 9,250 tons on 25 feet draught, and gross displacement of 13,600 tons. The engines, built by the same firm, have cylinders 30 inches, 49 inches and 78 inches, and 51 inches stroke, which are supplied with steam from three double-ended boilers, and will drive the vessel when fully loaded a speed of 10½ knots. She has been built to the order of Messrs. Crow, Rudolf & Co., of Liverpool, on the spar deck principle, with double bottom right fore-and-aft, and is intended for general trades.

Petroleum Tank Steamer America.—There was launched from the Yard of Messrs. Palmer's Shipbuilding and Iron Company, Jarrow-on-Tyne, England, on October 19th, the S. S. *America*, a large petroleum tank steamer built to the order of the American Petroleum Company. She was constructed under the specification and inspection of Messrs. Flannery and Blakiston, London. The vessel is 345 feet long between perpendiculars by 43 feet 10 inches beam, and 29 feet 1 inch moulded depth. The machinery is placed right aft. The cylinders are 27 inches, 44 inches and 72 inches in diameter by 48 inches stroke. Steam is supplied from two large double-ended boilers fitted with twelve Fox's furnaces of large size, which will give the ship a high speed at sea. The vessel is subdivided into sixteen tanks, with a total capacity of 4,500 tons of petroleum, water ballast being carried in the fore and after peaks, and in several tanks under the machinery suit-

able for storage of fresh water, and in addition she will be fitted with a powerful evaporator. The bunkers have a capacity of 900 tons of coal, sufficient for outward and homeward voyage. Two very powerful Worthington pumps are fitted in a pump room for rapid discharge of the oil cargo, and separate pumps deal with all water ballast. The ship is lighted throughout with electric light, with large sunlights for use on deck when in port. The installation is specially arranged to comply with the new United States Act with regard to electric light on petroleum steamers.

Whaleback Building on the Pacific.—There are people who believe—or profess to, at least—that with the total loss of the steamer *Charles W. Wetmore*, now ashore on the coast of the Pacific, there will be no further attempt to introduce that type of craft upon that ocean, or for that matter, upon any oeean. These disbelievers, however, forget that the whaleback steamer *Charles W. Wetmore* is not the only vessel of that kind upon salt water. They do not recollect, evidently, that the *Joseph L. Colby*, a whaleback counterpart of the *Wetmore*, has been for eighteen months upon the Atlantic coast, regularly running to different ports, and that she is just now bringing ore from the mines in south Cuba to the Maryland Steel Co. at Sparrow's Point, Md. They forget that this steamer has spent this long time in giving practical and visible evidence that whaleback vessels are adapted to coastwise and ocean service. It is so easy to forget the vessel that is doing well and so easy to see away across the continent the vessel of this type that is having hard luck—that has met with mishaps that would long ago have ended the careers of many vessels.

That the Pacific Ocean is not too severe for whalebacks is shown by the fact that at the enterprising town of Everett there are now being assembled the frames, plates and stringers that are to compose the first whaleback steamer built upon the Pacific coast, fitly called *Everett*, in honor of the town, at which has been established the shipyard of the Pacific Coast Barge Co., practically a part of the American Steel Barge Co., of West Superior, Wis.

This vessel is known as *No. 301*, at present. She will be 360

feet long, 42 feet beam, and 26½ feet deep, and will carry a cargo of 4,350 tons on a maximum draft of 19 feet. She is to be driven by a triple-expansion engine, whose cylinders have diameters of 24, 37 and 62 inches, and a stroke of 42 inches, to develop about 2,000 horse power and a speed of between 13 and 14 knots. Steam is to be supplied by four Scotch boilers, 11½ feet long by 11 feet in diameter, each having 2 furnaces, and to carry 160 pounds of steam. Her cargo from Puget Sound will consist of coal and lumber, and she is expected to carry back general package freight. She will have five large turrets and five masts, four of which will be set in small turrets along the main deck. Upon these turrets is built a deck that is cabined in the usual manner, there being twenty staterooms, containing accommodations for sixty passengers. The quarters for the crew are down in the hull aft, and are ample. She is expected to make the run from Everett to San Francisco, a distance of about 800 miles, in about 2½ days. As will be noted, she will be built to carry a large deck load of lumber, which is to be scientifically stowed. As usual with the whaleback steamers, this one will have a water ballast double bottom, supplemented by a tank at each end and one amidships, the whole aggregating a capacity of 1,800 tons Her scantlings are to be particularly heavy, and she will be expected to withstand almost anything on the Pacific except a lee shore.—"Seaboard."

OBITUARY.

ALEXANDER C. KIRK, LL. D.

Never in the history of marine engineering has our profession sustained a greater loss than by the death of Dr. Kirk, which occurred in the early days of October. Few among us can have failed to experience keenly that sense of regret which inevitably follows the removal of a great master; and to those to whom he was more than a name that regret must be deepened by the recollections of the benefits which never fail to be derived from personal intercourse with such a mind.

The importance of the early steps in the career of every man who is destined to rise to leadership in any profession cannot be overestimated, and all must recognize the wisdom which directed those of Dr. Kirk. On leaving school at Arbroath he was sent to Edinburgh University, at a time when a university education was little thought of as part of the training of an engineer. But to those who knew him the strong influence of the years spent at college was clearly present. There, under Forbes and Kelland, names both known in the world of science, he gained a strong grasp of scientific principles; and though many of the details may have evaporated in the exacting duties of his after life, his mode of thought often betrayed his earlier study of mathematics.

On leaving the University he applied to Messrs. R. Napier & Sons, of Glasgow, and after two years spent as a joiner, on the advice of Mr. Robert Napier, the founder of that firm, he was taken as an apprentice. There he came under the influence, not only of Robert Napier, but also of David Elder, of whose abilities and work he always spoke with the greatest respect. The firm was then attaining to that fame which it has so long and ably held, and to which no one has more fully contributed than Dr. Kirk himself. The marine engine was rapidly developing from its infancy, and nothing could be more inspiring than his

witnessing and probably taking part in many of its improvements.

On completing his apprenticeship he went for some years to London, to Messrs. Maudsley Sons & Field, where he quickly rose to a position of trust. His most important step was his appointment as engineer to the Paraffine Oil Works of Messrs. Young, Nuldrum and Binney, Bathgate. Soon he found ample scope for his abilities, as he was called on to design and construct new works at West Calder; and he used to say that then he did more original engineering than at any other period of his life. Here he made his famous experiments and inventions in connection with refrigerating machinery, and in 1861 applied an ether machine to the cooling of paraffine oil. This was the first application of refrigerating machinery to manufacturing purposes.

He next found his way back to Glasgow as manager of Messrs. James Aitkin & Co.'s Engine Works at Cranstonhill, and about 1870 was appointed engineering manager of Messrs. John Elder & Co., a post which he retained till 1877. In this position he did much lasting work for the improvement of the marine engine. Leaving aside minor details, it is not too much to say that the present form of the marine engine, as we see it in all the fast merchant steamers to-day, is largely due to the designs he carried out at this time. And the machinery for H. M. S. *Nelson* is the precursor of the light-framed engines so common in our war vessels now. It was a bold step in its day and well carried out. Above all must be mentioned his having, in 1874, fitted the first triple-expansion engine on board the *Propontis*, intended to work with 150 pounds steam pressure. The boilers, which were not made by his firm, were a failure, and due to this cause alone the success of the triple-expansion engine was not insured till in 1880 he induced Messrs. George Thompson & Co. to have one fitted on board the now famous *Aberdeen*. He was then senior partner of the firm of Messrs. Robert Napier & Sons, where he had started as an apprentice.

We have only been able to mention his more striking achievements. These are but an index of the restless energy of his mind, ever absorbed in perfecting some detail or inventing some

new process of manufacture. All who knew him—and we speak from knowledge gained by many years of work under his guidance, which will always leave a vivid memory of the man—were impressed with his powerful and rapid grasp of the central facts of all the problems which came before him—impatient always of mere refinements and side issues—the thoroughness of his work, and the patience with which he would reject scheme after scheme, always searching for some better solution of his difficulty, or some simpler method of construction, never sacrificing good work to mere commercial success. He is justly acknowledged the first marine engineer we have yet seen.

For many years he has been in failing bodily health, but to the last his mind retained its wonted power, till he was suddenly cut off in the midst of his work at the comparatively early age of sixty-two. Length of life is not to be measured by years in a case like his, and to-day the world is richer for his having lived.

JOHN H. MACALPINE.

CHARLES E. DE VALIN.

Chief Engineer Charles E. De Valin U. S. Navy, was born in Baltimore, Md., August 28, 1836. He entered the Navy as a Third Assistant Engineer Sept. 20, 1858, and was promoted to Second Assistant January 17, 1861; to First Assistant March 17, 1863, and to Chief Engineer March 21, 1870.

Nearly half of his career in the Navy was spent at sea, and he had served in all parts of the world. His last active duty was as Chief Engineer of the *Baltimore*.

His shore duty included service as an instructor at the Naval Academy and as Chief Engineer of the Washington Navy Yard. He had also spent some time in experimental work.

When ordered to the *Baltimore* he appeared in perfect health, but the cares and responsibilities incident to his duty with the powerful machinery of that vessel proved too much for him, and in the spring of 1891 he had a stroke of paralysis, from which he never wholly recovered. He was placed on the Retired List of the Navy April 11, 1892, and died at Washington only five days later, April 16.

BOOK REVIEW.

COMPRESSED AIR. (No. 106 Van Nostrand's Science Series.) The interest which has undoubtedly been aroused by the valuable article of Colonel Soliani, Chief Engineer in the Italian Navy, (published in the JOURNAL for May of this year), on the advantages of compressed air in the transmission of power on board ship, will lead to a desire for data and formulæ in a convenient form which will enable problems in connection with the use of compressed air to be solved.

The excellent little volume under discussion answers this demand in a handy form. It consists of a reprint of papers read before the Institution of Civil Engineers (England), by Professor A. B. W. Kennedy and Prof. W. Cawthorne Unwin. These papers contain data of tests of the Popp system of compressed air, in Paris, by the authors and by Prof. Riedler, the distinguished German engineer and professor at the Royal Polytechnic in Berlin. In addition, Prof. Unwin's paper gives all the formulæ necessary for calculations in connection with "compound" compressors, the most improved form.

The reputation of the distinguished authors is sufficient to guarantee the value of the book, but the formulæ have been tested by use and found to be entirely reliable.

Published by the D. Van Nostrand Co., 23 Murray street, New York. Price, 50 cents.